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Development of miniature pilot operated digital hydraulic valve

Department of Engineering Design and Production

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ABSTRACT OF
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This thesis describes the design, manufacturing and testing of a miniature hydraulic on/off valve. The valve has a solenoid actuated hydraulic pilot stage. The on/off valve is designed to be used as a part of a digital flow control unit (DFCU).

In this thesis is first designed a miniature solenoid actuator by using analytical equations and finite element simulations. A prototype of the actuator is built and tested. The designed actuator is used for actuating the pilot stage of the designed hydraulic valve. The design process of the hydraulic valve and simulations of the valve are presented. A DFCU prototype consisting of four designed on/off valves is built and tested.

The main parts of the DFCU prototype are four separate layers mounted together to form the necessary features for the magnetic circuits and the flow channels of the four valves. The physical size of a single on/off valve in the prototype is about 4,5 cm^3.

The maximum operating pressure of the designed valve exceeds 25 MPa. The opening response time of the valve is 1,3 to 1,6 ms and the closing response time between 1,4 and 2,7 ms depending on the operating pressure. The maximum flow rate of the valve is about 9 l/min with a pressure difference of 25 MPa.

The designed valve meets most of the requirements placed for it. It proves that pilot operation can enable further miniaturization of hydraulic on/off valves used in digital hydraulic applications.

Keywords:	digital hydraulics, digital valve, miniature valve, DFCU, magnetic circuit, solenoid
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Tämä diplomityö kuvaa pienikokoisen hydraulisen on/off venttiilin suunnittelun, rakentamisen ja testaamisen. Venttiilissä on solenoidikäyttöinen esiohjausventtiili ja se on suunniteltu käytettäväksi digitaalisen venttiilipaketin (Digital Flow Control Unit, DFCU) osana.

Työssä esitellään ensin pienikokoisen solenoiditoimilaitteen suunnittelu käyttäen analyyttisiä kaavoja sekä elementtimenetelmää. Toimilaitteesta rakennetaan prototyyppi ja se testataan. Suunniteltua solenoidia käytetään hydrauliventtiilin esiohjausventtiilin toimilaitteena. Hydrauliventtiilin suunnitteluprosessi ja venttiilin simuloinnit esitellään työssä. Työssä myös rakennetaan ja testataan neljä venttiiliä sisältävä venttiilipaketti.

Venttiilipaketti koostuu neljästä levymäisestä osasta, jotka yhdessä muodostavat toimilaitteiden magneettipiirit sekä venttiilien virtauskanavien osat. Yksittäisen suunnitellun miniatyyriventtiilin tilavuus venttiilipaketissa on noin 4,5 cm^3.

Suunniteltu venttiili toimii vielä yli 25 MPa paine-erolla. Sen avautumisvasteaika on 1,3 - 1,6 ms ja sulkeutumisvasteaika 1,4 - 2,7 ms. Venttiilin maksimivirtaukseksi mitattiin noin 9 l/min 25 MPa paine-erolla.

Suunniteltu venttiili täytti suurimman osan sille asetetuista suunnittelukriteereistä. Voidaan todeta, että esiohjaus mahdollistaa digitaaliventtiilien pienentämisen entisestään virtauskapasiteetin säilyessä hyvänä.

Asiasanat:	digitaalihydrauliikka, DFCU, solenoidi, esiohjattu
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Preface

I want to thank my instructor Jari Kostamo for giving me a great starting point in developing this valve as well as for innovative ideas and help. My supervisor Matti Pietola I want to thank for letting me work independently and trusting me to progress on my own. I also want to thank the rest of the staff in the laboratory of hydraulics at Aalto University, especially Jouni Pekkarinen for teaching me how to use a lathe and Esa Kostamo for expert help with the machining of the more difficult parts of the DFCU prototype.

These are interesting times for hydraulics and I believe that digital hydraulics will be a significant factor in developing hydraulics to a more environment friendly direction. It is interesting and motivating to work at a field which is developing as fast and is as promising as digital hydraulics.

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Tapio Lantela

Abbreviations and Symbols

CFD	Computational Fluid Dynamics
DAQ	Data Acquisition
DFCU	Digital Flow Control Unit
FEM	Finite Element Method
FEMM	Finite Element Method Magnetics (a program)
IHA	Department of Intelligent Hydraulics and Automation at Tampere University of Technology
PCM	Pulse Code Modulation
PNM	Pulse Number Modulation
PWM	Pulse Width Modulation

a	Acceleration [m/s^2]
A_a	Area where the magnetic flux passes between the armature and the core [m^2]
A_c	Cross sectional area of a magnetic flux path [m^2]
A_o	Area of an orifice [m^2]
A_p	Area affected by pressure in a poppet [m^2]
A_{piston}	Area of the piston [m^2]
A_w	Cross sectional area of a wire [m^2]
B	Magnetic flux density [T]
d	Diameter of a flow channel [m]
D_i	Inner diameter of the coil [m]
D_o	Diameter of an orifice [m]
D_p	Diameter of the piston [m]
f	Fill factor for the coil
F_m	Magnetic force [N]
F_p	Pressure force [N]
h_c	Height of the coil [m]
I	Current in the coil [A]
I_{max}	Maximum current in the coil [A]

l	Opening of a seat valve [m]
l_c	Length of a flow channel [m]
l_s	Length of a magnetic flux path [m]
l_m	Travel distance of the main valve's poppet i.e. opening of the main valve [m]
l_p	Travel distance of the armature i.e. opening of the pilot valve [m]
l_w	Length of the wire in the coil [m]
L	Inductance of the coil [H]
m	Mass [kg]
N	Number of loops in the coil
p	Pressure [Pa]
p_1	Pressure on the upper side of the orifice [Pa]
p_2	Pressure on the lower side of the orifice [Pa]
P	Heating power of the coil [W]
Q	Flow rate [m^3/s]
Q_p	Flow rate through the pilot valve [m^3/s]
R	Resistance of the coil [Ω]
R_m	Reluctance of a magnetic flux path [H]
R_{tot}	Total reluctance of the magnetic circuit [H]
V_p	Fluid volume displaced by the piston's movement [m^3]
U_m	Magnetomotive force [A]
U	Supply voltage for the coil [V]
w_c	Thickness of the coil [m]
α	Angle of the poppet's cone [rad]
η	Dynamic viscosity [Pa s]
μ	Flow coefficient
μ_0	Permeability of vacuum [H/m]
μ_r	Relative permeability
ρ	Density of fluid [kg/m^3]
ρ_c	Resistivity of copper [Ω m]
τ	Time constant of the coil

Contents

Chapter 1

Introduction

Hydraulic systems are widely used both in mobile and industrial applications. Compared to other options, hydraulic actuators have an outstanding power to size ratio, proven reliability and intrinsic overloading protection with pressure relief valves. Hydraulics, however, is not commonly thought of as a very energy efficient solution. Therefore, the current trend of energy efficiency has in many cases resulted in replacing hydraulic actuators and transmissions with mechanical or electric ones. It is true, that for decades hydraulic systems have been designed for maximum power and control performance and energy efficiency has been sacrificed in the process.

The current state of the art in hydraulic systems are load sensing systems with increasing amount of electrical sensing. Load sensing systems are energy efficient when using only one actuator but when simultaneously using multiple actuators with different pressure requirements the efficiency can be very low. One proposed solution to the energy efficiency as well as to several other problems is so called digital hydraulics. Digital hydraulics is not a clearly defined subset of hydraulics but in general it is considered to consist of solutions where actuators or valves have only a discrete number of operating states [18].

The main idea in digital hydraulic valves is to combine on/off valves with fixed orifices into valve packages and use them to control the flow rate of hydraulic fluid in a similar manner as normally is done with a proportional valve which has one variable sized orifice. Spool type proportional and servo valves are usually used in hydraulic systems where accurate position, velocity or pressure control of an actuator is required. Replacing proportional valves with digital hydraulic valves brings several benefits. A few of the most important benefits are energy savings, robustness and faster and more accurate response [20].

Currently there are thousands of different hydraulic valve types optimized for different tasks. Some examples of those are proportional directional valves, pressure relief valves and pressure reducing valves. Digital hydraulic valves offer the possibility to replace all the currently used hydraulic valve types with only a couple of valve types combined into intelligent valve packages. There are already examples where a valve designed for a specific application has been replaced with a digital valve [14]. In an ideal case the function of a digital valve package can be changed by merely changing the algorithm controlling the package. Therefore the complexity of the current hydraulic valves, which causes high cost, is transferred from the hardware to the software controlling the digital valve package. [20]

Digital hydraulics is a new field of research and there are no commercial products available yet. The currently available commercial on/off valves are not very suitable for digital hydraulic systems and therefore there is a need to develop smaller and faster on/off valves. The goal for this thesis is to develop an on/off valve which can be used to build a fast, robust and small digital hydraulic valve package. After some background information, this thesis presents the basic design of a miniature pilot operated on/off valve with the necessary calculations. Then the actuator of the pilot valve as well as the hydraulic parts of the valve are simulated. A prototype of the actuator and a four-valve digital valve package are built and their tests are also presented in this thesis. Some control electronics were also designed and built for controlling the designed valve but more detailed discussion about them is outside the scope of this thesis.

Much of the research involved in this thesis has already been introduced in a conference article published earlier [15]. This thesis however discusses the research in more detail. No references to this article will be used further in the thesis.

Chapter 2

Digital hydraulic valves

2.1 Digital flow control methods

There are several different methods for utilizing on/off valves in controlling an actuator. Simple hydraulic circuits demonstrating three of these methods are displayed in figure 2.1. The most basic type is so called bang-bang method where a single on/off valve is held open until the actuator has reached the desired position and after that the valve is closed. This method requires only one valve and the valve is also switched on and off only once during the movement of the actuator. However, the maximum speed of the actuator is limited by the flow capacity of the single valve and the accuracy of positioning depends on the flow rate through the valve. If the actuator moves very fast the positioning is more difficult. Sudden stopping of a fast moving actuator also creates pressure shocks in the system due to the inertia of the actuator and the fluid. The bang-bang method requires only simple hardware but the control performance is poor and that is why it is not used often. [20]

The bang-bang method can be improved in at least two ways. One way to improve it is to cycle the valve rapidly between open and closed states. This way can be created an average flow rate between zero flow and the maximum flow rate of the valve. The most popular method for this is Pulse Width Modulation (PWM) which is analogous to controlling current with PWM in electronics. In hydraulic PWM the on/off valve is switched with a constant frequency between open and closed states and the duty cycle is varied to change the flow rate. The frequency needed for sufficient control performance depends on the application but a typical frequency is 50 Hz [18]. Only one valve is needed for PWM control which makes the system simple. However, if the desired flow rate through the

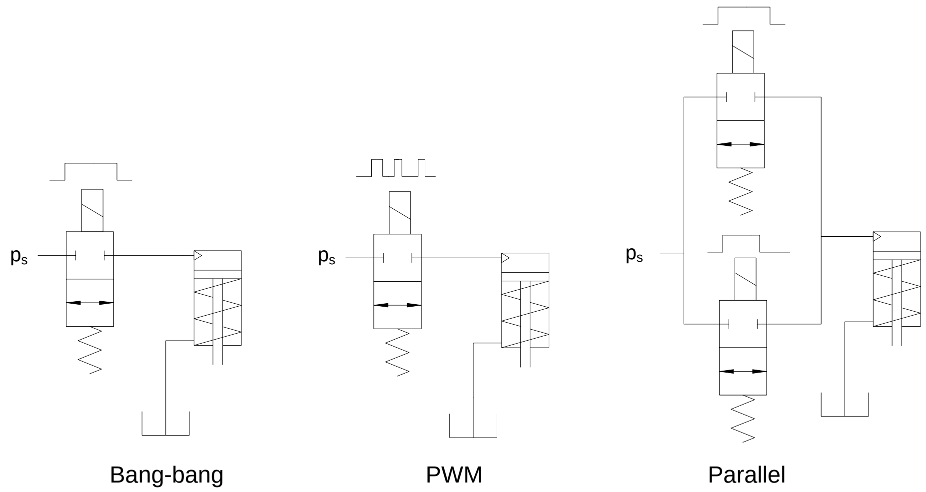


Figure 2.1: Digital hydraulic methods for controlling flow.

valve is not zero or maximum flow rate of the valve, the valve needs to be switched continuously to maintain the desired flow rate. Thus the valve has to endure a large number of operating cycles in quite a short time which demands excellent fatigue life from the valve. [20]

Another way to improve the bang-bang method is to add a second smaller valve parallel to the first valve. This way the larger valve provides the large flow required for the fast movement of the actuator and the smaller valve increases the speed slightly. The larger valve can be switched off when the actuator is close to the target position. After that the speed of the actuator is determined by the flow through the smaller valve, which slows down the actuator, making positioning more accurate. Therefore connecting valves in parallel increases both the maximum speed of the actuator and the positioning accuracy. It also decreases pressure shocks since the actuator does not stop at once. A price to pay is the added complexity of the system from adding a second valve.

2.2 Digital flow control unit

A Digital Flow Control Unit (DFCU) consists of several parallel connected on/off valves which together create one variable sized orifice. Thus a DFCU can be used for example as one control edge in a directional valve. The drawing symbol and the hydraulic circuit of a DFCU are displayed in figure 2.2. A valve package consisting of several DFCUs is called a digital valve system [19].

The idea of using several on/off valves connected in parallel for controlling flow is over a hundred years old but until recently it has not been much applied in practice [20]. The idea was brought back to current research about ten years ago at the Department of Intelligent Hydraulics and Automation (IHA) in Tampere University of Technology (TUT). DFCUs have been researched intensively at TUT for the past years and they have also been applied to industrial applications successfully [6]. Parallel connected valves and the digitalization of hydraulics has been compared to digital revolution in electronics [19]. A quite similar revolution is however not to be expected in hydraulics since contamination of the hydraulic fluid and properties of laminar flow limit the miniaturization of the components. Current manufacturing and materials technology have however already made it possible to build valves consisting of a large number of miniaturized on/off valves.

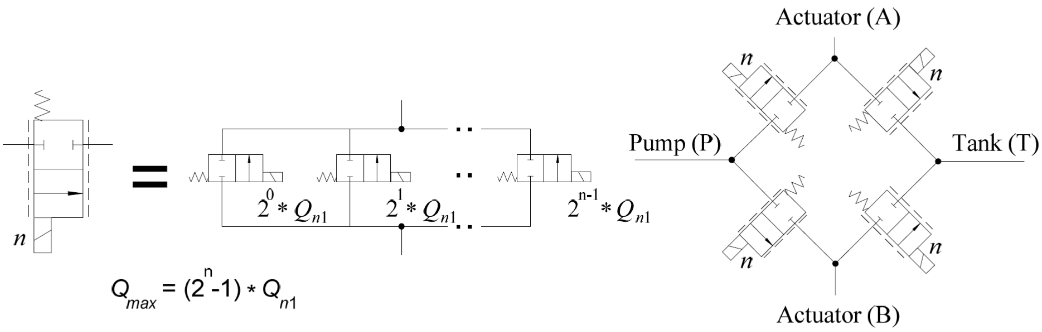


Figure 2.2: The drawing symbol and the hydraulic circuit of a DFCU and the hydraulic circuit of a digital hydraulic valve system. [26]

DFCUs are usually built with seat type valves which are generally considered leak-free as opposed to spool type valves. The structures of a seat and spool type valves are displayed in figure 2.3. A seat type valve closes one flow channel i.e. one control edge of the valve by pressing a poppet against a seat. A spool type valve can instead control several control edges simultaneously with a spool moving inside a bore. Thus for example a 4/2 proportional directional valve can be built with only one spool but building it with seat type valves requires four valves. However, with one spool the control edges cannot be controlled independently from each other, which limits the performance of the system. For example cavitation of the valve can be prevented by controlling the inflow and outflow of the actuator separately [20].

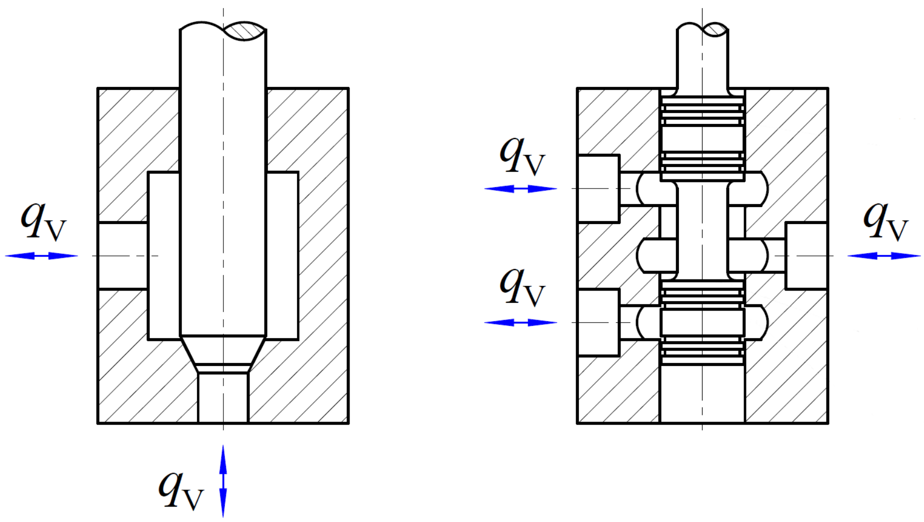


Figure 2.3: The structure of a seat type on/off (2/2) valve and a 3/2 spool type directional valve. [11]

Currently the fastest spool type proportional or servo valves have response times of around 5 ms but the response time depends on the amplitude of the spool's movement. Another problem with proportional valves without position feedback is also the zero point drifting. The response time of an on/off valve is always more or less deterministic since the valve only has two discrete states and the moving part in the valve moves against a mechanical stop. Also because in a DFCU all of the valves can open simultaneously, the response of a DFCU is not dependent on the amplitude of the required change in flow rate [19].

Digital hydraulics is still a new and not widely used technology. There are currently no digital hydraulic valve packages commercially available even though the technology has been proven to work both in research laboratories and in industrial applications. The current digital hydraulic solutions have been realized with separate commercial on/off valves mounted to a large manifold. An example of a DFCU which has been used in research projects is displayed in figure 2.4. The current commercial on/off valves are not designed for this kind of applications and therefore they are quite large and their response time is slow, in the order of 10 ms with proper control electronics. This is why the research effort in Aalto University and Tampere University of Technology is aimed at developing technology that would enable Finnish industry to commercialize digital hydraulic technology more widely than currently. This thesis is part of that

Figure 2.4: A four-way DFCU consisting of commercial Hydac WS08W-01 cartridge valves. Each control edge has five valves. Dimensions of the DFCU are 150 x 200 x 240 mm. [31]

research.

There are several prototypes of digital valves built in research laboratories. On/off valves specially designed for DFCUs have been developed at IHA for some years now. First, a bistable valve was designed [29] and after that a refined version of it called the hammer valve [30]. The hammer valve's response time is about 2 ms, it has a good flow rate to size ratio and it does not require power to maintain an open or closed state. Downside of the bistable valve prototype was its difficult to manufacture pressure compensated poppet. At IHA was also built a 16-valve prototype DFCU with the hammer valves [31]. Recently, the research at IHA has concentrated in valves actuated by regular solenoids. The latest valve introduced by IHA is a miniature needle valve with a 10 mm diameter [9] [10]. The Proto$\varnothing$10 has a response time of about 1,5 ms and its size is very small. Due to the small size and the direct operated structure also its flow capacity is quite small.

2.3 Modulation types

The term modulation is used to describe how the state of a DFCU, i.e. the configuration of open and closed valves, is derived from the desired flow rate. The orifices in the on/off valves of a DFCU may all be of similar size or they all may be different to achieve certain benefits. The modulation where all the orifices have the same size and therefore the same flow capacity is called Pulse Number Modulation (PNM). In other words the flow rate is determined by the number of open valves in the DFCU. The modulation where there are different sized orifices in the valve is called Pulse Code Modulation (PCM). Thus the flow rate of a pulse code modulated DFCU depends on the number of open valves and the coding used to determine their orifice sizes. Currently the most popular modulation in DFCUs is PCM where the orifice sizes are coded with binary coding. This means that the size of the smallest orifice is determined by the desired resolution of the DFCU. The rest of the orifices are always twice as large as the one step smaller orifice (1, 2, 4, 8, 16 etc.). Resolution in digital hydraulic valves means the amount of the smallest change in the flow rate which can be caused by changing the valve's state. [19]

In theory binary coding makes it possible to achieve the best resolution with a certain number of valves because n number of valves can be combined to create 2^n different flow rates. The valves used in a binary coded DFCU can all be similar but their flow can be restricted by using a binary coded orifice plate in series with a valve. Smaller valves can also be used for controlling the smaller orifices to save space or costs [19]. On the left in figure 2.5 is displayed the flow rate of a proportional valve with respect to the opening of the valve. The resulting flow rate is clearly nonlinear and there is some uncertainty in it. In the middle of figure 2.5 is displayed the flow rate of a 5-bit binary coded DFCU. A 5-bit DFCU has $2^5 = 32$ different states which can create 32 evenly distributed flow rates with a certain pressure difference. On the right is the flow rate of a 7-bit DFCU. A 7-bit DFCU has $2^7 = 128$ different states and therefore it gives about the same controllability as a normal servo valve [20].

The binary coding however has its downsides, namely pressure shocks created by insufficiently accurate timing of simultaneous valve opening and closing. For example when the flow rate is changed from half of the DFCU's flow capacity to less than half, the largest valve has to be closed and the rest of the valves opened. If the opening and closing events are not perfectly simultaneous there may be a moment when all the valves are closed at the same time, which may cause a pressure shock in the valve.

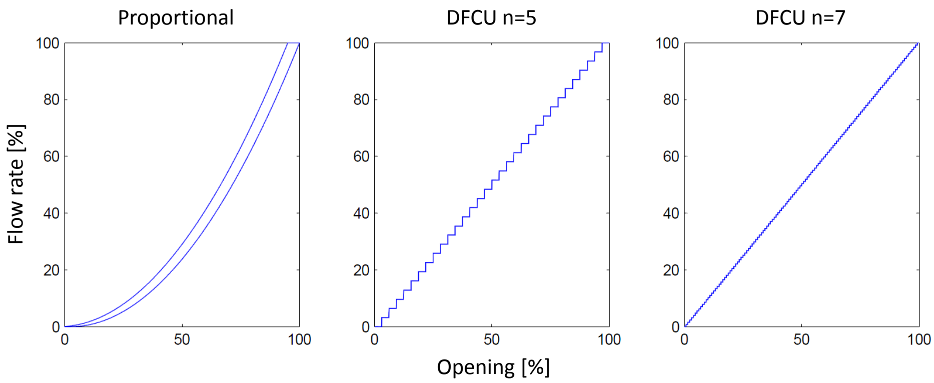


Figure 2.5: The flow rates with respect to the opening of the valve in a proportional spool valve and a binary coded DFCU with five or seven valves. [8]

This phenomenon is illustrated in figure 2.6. [19] Also if the best resolution of a binary coded DFCU is required at any flow rate, the smallest valve will be switched much more often than the largest valve. This leads to uneven stressing of the valves which may lead to a sooner failure of the smallest valve. The full resolution is however not always necessary with larger openings of the valve [19].

Another modulation type is where the valve orifice sizes are coded according to the Fibonacci series. In the Fibonacci series the following number is always the sum of the two previous numbers (1, 1, 2, 3, 5 etc.). When comparing binary and Fibonacci coded DFCUs with the same number of valves and the same resolution, the Fibonacci coded DFCU will have a smaller flow capacity. However, Fibonacci coding brings redundancy to the DFCU since the desired flow rate can always be created with at least two different combinations of opened valves. This also reduces the number of valve switchings needed if the controller is optimized properly. [13]

Pulse number modulation is very different from binary modulation where a small resolution and a large flow capacity are achieved with a small number of valves. In a PNM DFCU all the valves have orifices of the same size which is determined by the minimum desired resolution of the DFCU. Valves are never opened and closed simultaneously which reduces pressure shocks significantly. Because the valves are all similar, it also does not matter which valve is opened when more flow is needed. This makes it possible to stress all the valves evenly which increases the DFCU's fault tolerance in the long run. If one or multiple valves break down so that

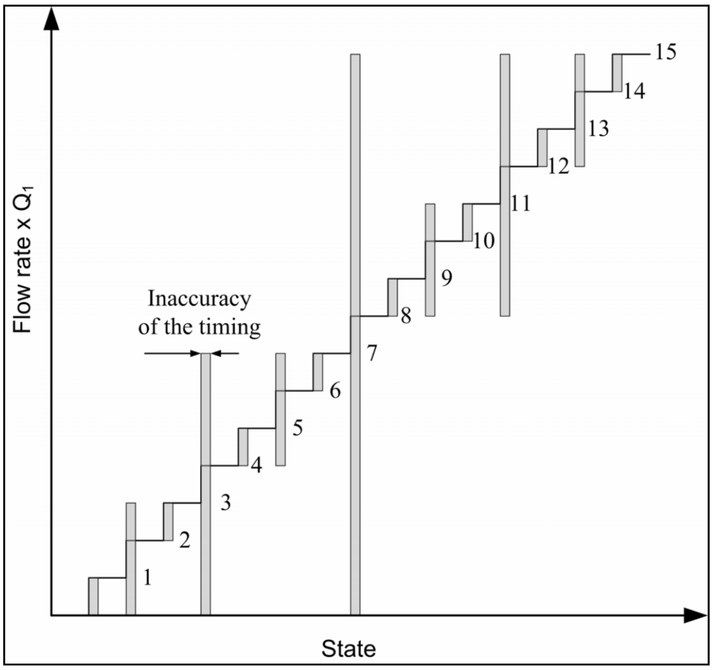


Figure 2.6: State uncertainty in a binary coded DFCU caused by inaccuracy in the opening and closing times of the valves. [19]

they do not open anymore it has no effect on the resolution of the DFCU, but only on the maximum flow capacity. Downside is that to achieve the same resolution as for example a binary coded DFCU with seven valves, a PNM coded DFCU must have 127 valves. [18]

There are also several modifications and combinations to the previously mentioned codings. For example the flow capacity of the Fibonacci coding can be increased by replacing one of the smallest valves with a larger one (1, 2, 3, 5, 8) [13]. It is also possible to combine PNM and binary coding by making a DFCU which mainly uses PNM coding but has some smaller binary coded valves for increased resolution (for example 1, 2, 4, 8, 8, 8). It is also possible to increase the resolution of a DFCU by controlling some of the valves with PWM. If the PWM controlled valve has a small orifice, the resolution of a DFCU can be improved significantly without the pressure shock problems often associated with PWM control. [7]

2.4 Robustness and fault tolerance

Fault tolerance of DFCUs consisting of parallel connected on/off valves will be a significant improvement over traditional proportional valves. The simple structure of on/off valves makes them more tolerant of contaminated hydraulic fluid than proportional valves. The amount and direction of flow through a spool type proportional valve depend on the position of a spool moving inside a highly accurately machined bore in the valve. High machining accuracy is required for minimizing the gap between the bore and the spool because the larger the gap is the more fluid leaks between them. Leaking is an unwanted phenomenon because it increases energy losses and causes drifting of actuators and a decrease in control performance. The very small gap causes problems since contaminant particles may get stuck in the gap and prevent the spool from moving. The seat valves used in DFCUs do not suffer from similar problems.

Digital hydraulic valves are excellent especially in water hydraulics where the tight tolerance between the spool and the bore is even more problematic than in oil hydraulics. Water is less viscous than oil which increases leaking, but water is also a less effective lubricant than oil which increases friction wear and failure probability of the valves. This is why water hydraulic proportional valves are very expensive [20].

Fault tolerance is also improved due to the fact that there are multiple parallel connected valves in the same package. This means that even though it is more probable that one of the valves in a DFCU fails compared to the failure of a single proportional valve, it is very improbable that all of the valves in the DFCU fail at the same time. Therefore the hydraulic system controlled by a DFCU can remain operational even though some valves fail and in many cases failures do not even affect the performance of the system. Fault tolerance of a DFCU depends on the modulation, electronics and control algorithms of the DFCU.

A valve can fail in several different ways of which some are more difficult to compensate than others. Before a fault can be compensated, it has to be detected first. The best way to detect a fault would be to measure the position of the valve's poppet. This however may require modifications to the structure of the valve and also adding extra sensors. Other ways to detect faults are to measure the current in the solenoid's coil or use pressure sensors which often are already installed in the system. With proper sensors, the fault can be detected during the system's normal operation but often it is easier to implement a separate test sequence, which is run when the system is offline. When the faults are detected, they can often be compensated by automatically reconfiguring the controller of the DFCU . [26]

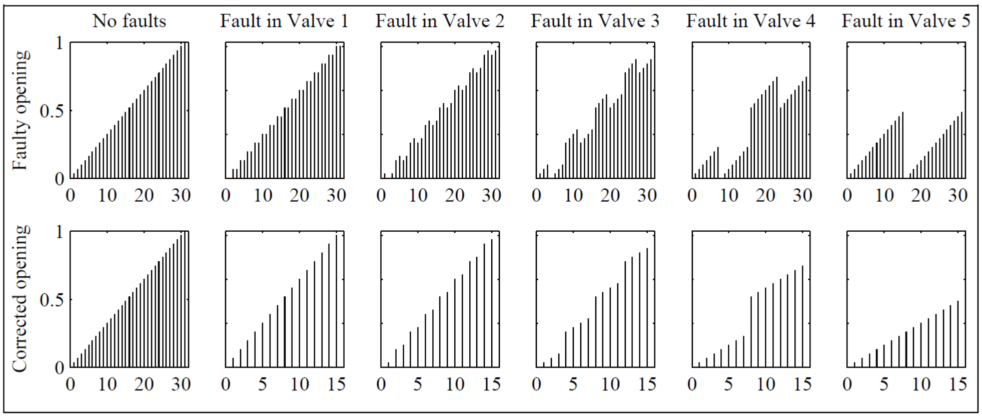


Figure 2.7: Compensation of a single jammed valve in a binary coded DFCU. [26]

In figure 2.7 is displayed an example of how a failure in a single valve, which is jammed in closed position, can be compensated in a binary coded DFCU. From the image can be seen that a DFCU's performance is not significantly reduced by a failure in some of the smaller valves as long as the failure is detected and compensated properly. The effect of valve failures is even smaller with Fibonacci coded or PNM coded DFCUs. [26]

2.5 Scaling laws

In order for DFCUS to be able to replace the currently used proportional valves, they need to be physically of the same size or preferably smaller than the proportional valves. This requires miniaturization of the DFCUs and therefore also miniaturization of the on/off valves used to build a DFCU. However, the physical dimensions of a DFCU are not the only reason to pursue the miniaturization of the on/off valves but miniaturization also brings other benefits.

Let's assume that all the dimensions of a seat valve are proportional to its opening l. When the opening is reduced, also the diameter of the valve's orifice is reduced and thus the area of the orifice A_o reduces exponentially compared to l i.e. $A_o \propto l^2$. The pressure force acting on the poppet is proportional to the area A_o. Therefore also the pressure force reduces exponentially as the opening reduces. This leads to that a smaller actuator is required to open or close the valve. If we assume that also the outer dimensions of the valve are proportional to the opening, the volume V of the valve is proportional to l^3. This applies also to the moving parts inside the valve which reduces their mass and therefore also improves the response time of the valve. [19]

Reducing the opening of the valve improves the response time because of the reduced travel distance of the poppet and also due to the reduced masses. The flow rate Q though the valve is proportional to the area of the orifice and thus it is reduced with a factor l^2. Therefore replacing a large valve with four smaller valves with $l = 0,5$ leads to the same flow rate as before ($Q_2 = 4Q_1l^2 = Q_1$) but with half the volume ($V_2 = 4V_1l^3 = 0,5V_1$). Thus miniaturizing valves not only reduces the volume of a DFCU while maintaining the same flow capacity but also improves response time. [19]

An added benefit is that the energy consumption decreases with smaller valves. The work W done when opening a valve is the product of the force and the opening i.e. $W \propto l^2 * l = l^3$. Thus the power consumption of the DFCU is decreased and therefore the control electronics can be cheaper and the issues with heating are reduced.

2.6 Actuator selection

The purpose of the valve actuator is to generate the necessary movement to open or close the orifice in the valve. Movement can be created with a number of methods ranging from electric current to controlled explosions. Electromagnetic devices are the most popular valve actuators but nowadays there are also a range of interesting active materials which could be used as actuators.

Active materials are materials that can change their shape when they are given an appropriate stimulus. The stimulus can be for example electric field, current, magnetic field, heat, light, a chemical compound etc. Many of the active materials are still used mainly in research laboratories but at least piezoelectric crystals are already widely used in industrial applications.

Piezoelectric crystals change their shape when an electric field is applied to the material. Piezos are very good for accurate positioning but it is not a requirement for the actuator of an on/off valve. The response time of piezo actuators is very good and their energy consumption is low. They can create a very large force compared to their size but the elongation of the crystal is only in the order of 0,1 % of its original length. This means that the force needed to open a small valve could be generated with as small a crystal as 1 mm x 1 mm but even with a length of 20 mm the elongation would be about 0,02 mm which is too little for a valve actuator. The high force and a small movement of a piezo can be converted to a lower force with a larger movement with simple lever systems or a small hydraulic piston. However in a miniature valve these systems take precious space and a large proportion of the generated movement is often lost because of the elasticity of the lever or the compressibility of the hydraulic fluid. [25]

Another interesting group of active materials is Magnetic Shape Memory Alloys (MSMA). MSMAs are metals which change their crystal structure under magnetic field. The change in the crystal structure creates an elongation of up to 10 % in the material and a quite high force. There are however two major drawbacks with MSMAs. The first is that the fatigue life of the actuator is questionable especially with larger strains. However fatigue tests have been successfully made with up to $2*10^9$ cycles with a 2 % strain [1]. The second and the more important drawback is that the magnetic circuit used to generate the necessary magnetic field is difficult to miniaturize which makes the whole actuator structure too bulky for a miniature valve [23].

Other active materials are for example magnetostrictive materials, electrostrictive materials, magnetorheologic fluids etc. but all of them require either more research for them to be used as valve actuators or have some property preventing their effective use. They are also usually expensive compared to electromagnetic actuators and difficult to acquire.

So we are back to the traditional electromagnetic actuators. The most used actuator device in valves is a unidirectional solenoid with spring return. Properties of a solenoid actuator are explained in detail in chapter 3.3. Many electromagnetic devices like voice coils and proportional magnets create a force or displacement which is proportional to the excitation of the coil in them but these proportional properties are not necessary in an actuator of an on/off valve. A solenoid can also be made bidirectional but this makes the structure of the solenoid more complex and difficult to manufacture [30]. Bidirectionality also does not bring much benefit if the power needed to keep the valve in open state with unidirectional solenoid is low enough.

The force generated by a solenoid is strongly dependent on the travel distance of the armature. If the movement of the actuator i.e. the opening of the valve is reduced to half while keeping the input power to the solenoid constant, the force generated by the solenoid increases to fourfold. This is favorable to miniaturization since in a smaller valve the opening is also smaller so the solenoid can either generate more force or input power can be reduced. Thus it can be concluded that even though active materials such as piezoelectric crystals could be used as an actuator for an on/off valve they do not bring much benefit when compared to a solenoid actuator.

The actuator selection for this thesis was limited to options other than a direct operated solenoid actuator. Since a solenoid still seems to be an excellent choice for the actuator of a miniature valve, the actuator for the valve designed in this thesis was chosen to be a solenoid with a hydraulic pilot stage. This way a very small solenoid is sufficient and the excellent power density of a hydraulic actuator enables further miniaturization of the valve.

Chapter 3

Theory and design

3.1 Main valve

The numeric design goals for the valve developed in this thesis are the following:

- 20 MPa maximum operating pressure
- 2 ms response time
- 1,4 l/min flow rate @ 0,5 MPa pressure difference
- Size less than 5 cm^3

In addition there are qualitative requirements i.e. the valve should be:

- Leak-free
- Preferably bidirectional
- Easy to manufacture
- Robust

Because the designed valve is required to be leak-free, it will have to be a poppet type valve. The poppet and the seat in a valve are usually axial symmetric for easy machining and also the orifice in the valve is circular.

Simplified geometry of a poppet valve is displayed in figure 3.1. The flow rate Q of turbulent flow through a circular orifice can be calculated with equation (3.1) where μ stands for the flow coefficient, A_o for the area of the orifice, p_1 for the pressure on the high pressure side of the orifice, p_2 for the pressure on the low pressure side and ρ for the density of the fluid [12]. In a circular orifice the area can be calculated from the diameter of the orifice D_o.

$$Q = \mu A_o \sqrt{\frac{2(p_1 - p_2)}{\rho}} = \mu\pi(\frac{D_o}{2})^2 \sqrt{\frac{2(p_1 - p_2)}{\rho}} \tag{3.1}$$

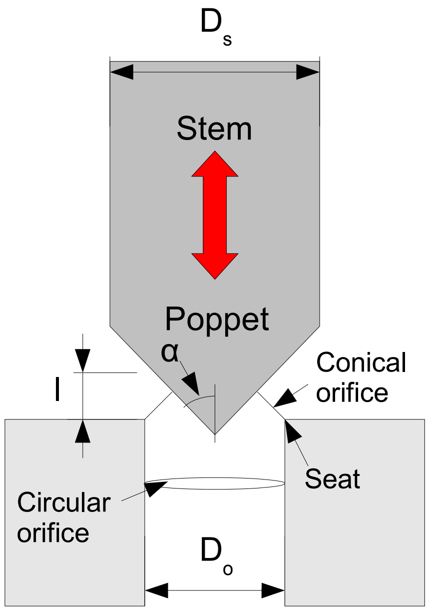


Figure 3.1: Simplified geometry of a poppet valve.

However the valve is not only a circular orifice since the poppet stays very close to the seat even when the valve is fully open. The minimum cross sectional area of the flow creates a conical surface between the seat and the poppet. If the surface area is flattened it forms basically a rectangular orifice. The height of the orifice is the distance between the poppet and the seat, and the width is approximately the circumference of the circular orifice. Flow through a rectangular orifice can be calculated with equation (3.1) by modifying the area term. By assuming that the area of the rectangular orifice is the product of circumference of the circular orifice and the distance between the poppet and the seat we get equation

(3.2) where l is the vertical opening of the valve and α is the angle in the poppet's cone.

$$Q = \mu \pi D_o l \sin\alpha \sqrt{\frac{2(p_1 - p_2)}{\rho}} \tag{3.2}$$

With a similar orifice diameter and a large enough opening the equation (3.2) gives a larger flow rate than equation (3.1). However there are basically both orifices present in the valve at the same time so when designing the geometry, the equation which gave the smaller flow rate was always used. The calculated flow rates are directly proportional to the flow coefficient μ. The flow coefficient is difficult to determine without Computational Fluid Dynamics (CFD) simulations or empirical tests so a quite conservative value of 0,6 was used in the calculations.

The required flow rate for the designed valve was 1,4 l/min with a 0,5 MPa pressure difference. With the equation (3.1) can be calculated that a circular orifice with about 1,2 mm diameter results to the desired flow rate. A density of 890 kg/m^3 was used for the hydraulic fluid in the calculations. Similarly with equation (3.2) can be calculated that the same orifice with a 0,45 mm opening and $\alpha = 45°$ passes the same flow rate. On the other hand the equation (3.2) can give infinitely large flow rates when the opening is increased which obviously is not possible for a seat valve. It is difficult to define when either of these equations give valid results and therefore an orifice diameter of 1,35 mm was chosen for the prototype. This was hoped to give some safety factor with the vague flow rate calculations.

Now that we have calculated the diameter of the orifice we can also calculate the force the pressure exerts on the poppet when the valve is closed. When pressure is considered evenly distributed on a surface, the force F_p exerted by it on any object can be calculated with equation (3.3) where Δp stands for the pressure difference across the object and A_p for the area of the object.

$$F_p = \Delta p A_p. = \Delta p \pi (\frac{D_o}{2})^2 \tag{3.3}$$

With the required maximum operating pressure on the other side of the poppet and the tank pressure on the other side there is a 20 MPa pressure difference over the poppet of the main valve. With equation (3.3) can be calculated that a 20 MPa pressure acting on the area of 1,35 mm diameter orifice causes a force of about 29 N to the poppet. In a poppet valve the pressure difference can be in either direction across the valve i.e. the pressure force can either try to open the valve or push the poppet against the seat even harder. Therefore if the valve is required to be bidirectional, meaning it should open and close normally regardless of the direction of the pressure difference, the valve's actuator needs to produce both pulling and pushing force.

3.2 Pilot stage

3.2.1 Structure

In effect the pilot stage is a differential hydraulic cylinder, which moves the poppet of the main valve and is controlled by the pilot valve. Hydraulic circuits of four different ways to realize a pilot stage are displayed in figure 3.2. To control a normal cylinder in both directions, both chambers of the cylinder must be possible to connect to the tank and the pressure channels. This would require a 4/2 valve. Such a valve with two different positions and four ports is quite complicated and difficult to miniaturize. On the other hand if the cylinder has a return spring, only one chamber needs to be controlled and therefore only a 3/2 valve with pressure and tank lines and one outlet line is required. A 3/2 valve is much easier to realize than a 4/2 valve.

When actuating the cylinder with a 3/2 valve, the controllable chamber is connected to the pressure line and the pressure pushes the piston in the cylinder to the other end of the stroke. When the chamber is disconnected from the pressure line and connected to the tank by the 3/2 valve, the return spring pushes the fluid from the chamber to the tank line and returns the piston to normal position. The side of the spring should be chosen so that the spring closes the valve i.e. the valve is normally closed. If desired, the 3/2 valve could be replaced by two 2/2 valves. One of the valves would connect the chamber to the pilot pressure and the other to tank pressure.

The system could be simplified even more by replacing the 3/2 valve with only one on/off valve, which connects the cylinder's chamber to the pressure supply. In this case there would have to be a throttle constantly open from the chamber to the tank line. This way the return spring can push the fluid from the chamber to the tank through the throttle when the connection to the pressure line is closed. In this case however there would also be a constant flow from the pressure line to the tank line while the cylinder is actuated i.e. the main valve is open. The throttle would have

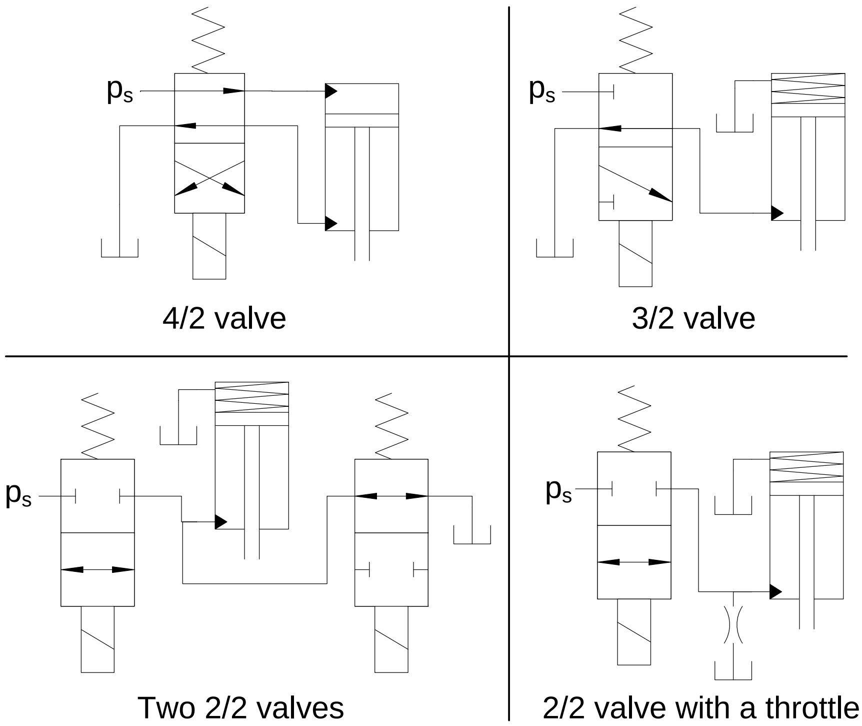


Figure 3.2: Hydraulic circuits of four different ways to realize the pilot stage.

to be very small in order to keep the flow rate small but this also would slow down the closing movement of the cylinder significantly. Both the opening and the closing cycles of the main valve should be equally fast for good control performance and there is no point in draining fluid from pressure inlet straight to the tank so this mechanically somewhat simpler method was rejected.

The pilot valve does not necessary have to be leak-free like the main valve so it could be either a spool or a poppet type valve. In this thesis it was selected to be poppet type since in this case the pilot valve was also desired to be leak-free.

In chapter 2.6 a solenoid with a return spring was chosen as the actuator for the pilot stage. In a 3/2 poppet valve two orifices need to be closed or opened and our actuator produces only linear bidirectional movement. This means that unless there are some kind of lever systems in the valve, the orifices need to be on the same axis as the stem of the solenoid's armature. Therefore the stem needs to go through the upper orifice. Simplified

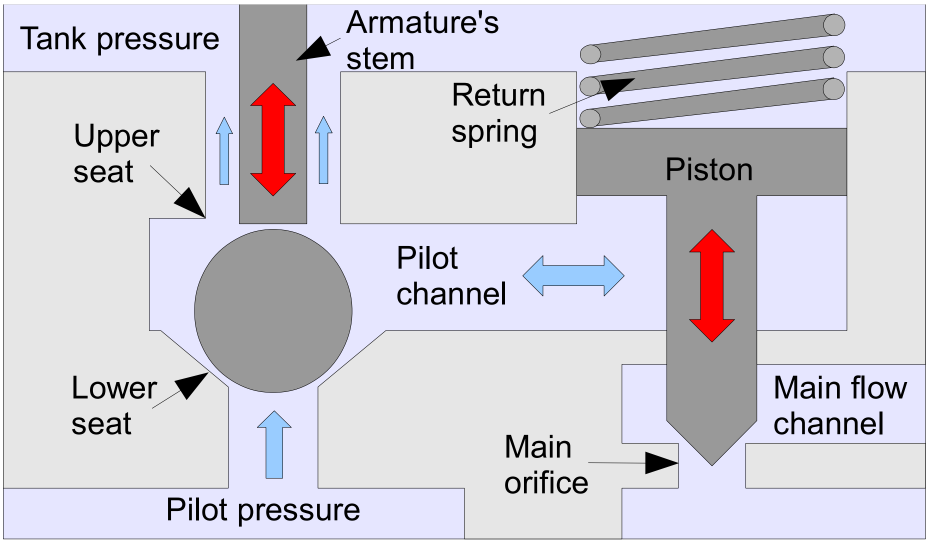


Figure 3.3: Simplified geometry of the hydraulic part of the valve. Pilot stage on the left and main stage on the right.

geometry of the hydraulic part of the valve is displayed in figure 3.3.

The downside of the fact that the stem moves through the upper orifice is that the stem reduces the orifice's area available for fluid flow. Therefore the diameter of the upper orifice should be larger than the area of the lower orifice in order for the flow areas and thus the flow rates for them to be similar. A larger diameter increases the pressure force subjected to the sealing element when the upper orifice is closed. Also either the upper or lower seat has to be chamfered to keep the ball trapped between the seats.

To seal the two orifices there either needs to be two separate poppet surfaces machined to the stem, or a separate sealing element which is moved with the stem could be placed between the two seats. In a miniaturized valve it is much easier to use a separate sealing element instead of manufacturing a stem with two poppet surfaces so it was decided to use a small bearing ball as the sealing element. This ensures a very good surface finish, tight tolerances and a hard material which is unlikely to deform in use. The bearing ball is not forced to the upper position when sealing the upper orifice but instead the flow from the pressure line through the lower orifice and the pressure difference over the ball lock it to the upper position.

3.2.2 Pressure, force and flow calculations

There are two ways to produce the necessary pressure difference to move the piston in the pilot stage. One way is to use the pressure difference over the main valve in the main lines. This however has some downsides. First, the pressure can vary a lot which makes it very hard to optimize the pilot stage as it works optimally with only one pressure level. If the maximum operating pressure of the valve is very high, the orifices of the pilot valve need to be very small in order to keep the maximum pressure force low enough so that the actuator can handle it. This consequently reduces the maximum flow rate of the pilot valve thus increasing the response time of the main valve. Second, there may be a situation where there is no pressure difference between the main lines. For example, when the flow to a cylinder chamber is controlled with the main valve and the cylinder is moved to the end of its stroke so that the pressure in the chamber is the same as in the supply line. In this situation there may not be any pressure difference to be used as pilot pressure.

The second option is to assume that there is a constant pressure supply available for the pilot stage or that it can be easily created. This way the pilot stage can be optimized for the selected constant pressure level and there are no uncertain situations with no pilot pressure. In the beginning of the design phase the necessary pilot pressure was expected to be easily available in the hydraulic system. A 10 MPa pressure level was selected as the pilot pressure.

Later while already prototyping this presumption was questioned and it was realized that the pilot pressure may not be easily available. For example in a load sensing system the standby pressure is usually between 1 and 3 MPa which is not enough for the designed pilot stage to function properly. This pressure level is also too low for making a fast and strong enough pilot stage with the design presented in this thesis. There are however simple and compact devices which can raise the pressure level high enough for the pilot stage to work effectively [22].

Properties of the solenoid actuator are described in more detail in chapter 3.3 but since the designing of the pilot stage and the actuator are tightly related they cannot be described totally separately. Therefore without further explanation in this chapter we assume that due to the requirements of the actuator the opening of the pilot valve should be as small as possible and also that the actuator can generate a bidirectional 5 N force. The reducing of the opening of the pilot stage is limited by the subsequent reduction in the flow capacity of the pilot stage but there are also two other important limiting factors.

The more important of these two are the contaminants in the hydraulic fluid. If the opening of the valve is small enough the contaminants, for example small metal chips, may get stuck between the seat and the poppet when the valve is open. The size of the particles in the fluid depends on the filtering class of the hydraulic system. Generally in hydraulic systems with servo valves particles larger than 5 μm are expected to be filtered out from the fluid. With spool type proportional valves the limit is around 10 μm and with pumps and cylinders from 10 to 20 μm [24, p. 185]. Since a digital hydraulic system is expected be more robust than a system with spool type proportional valves it is a good target that the digital valves would not require any higher filtering standards than the other components in the system.

So if the largest contaminant particles in the hydraulic system are expected to be from 10 to 20 μm in diameter, how small can the opening of the valve be? The relationship between the smallest orifice in a valve and the filtration class in a pneumatic system was discussed by Heikkilä [5]. Heikkilä studied the ratio the between the size of the orifice and the size of the allowed contaminants in commercial seat type pneumatic valves. The ratio varied between 4,5 to 42 but for the valves with an orifice diameter of under 1 mm the average ratio was about 13. Therefore the smallest opening of the pilot valve resulting in reliable operation could be somewhere between 0,13 to 0,26 mm.

Another factor limiting the miniaturization is the tightening manufacturing tolerances as the dimensions and the movements in the structure get smaller. If the opening of the pilot valve would be for example 0,1 mm and there would be a $\pm$0,03 mm manufacturing tolerance in the part defining the opening, there would already be a 30 % uncertainty in the real opening. This could significantly affect the response time of the valve or even prevent the valve from functioning correctly. Due to the impurities and the tolerance issues the minimum usable opening of the pilot stage was decided to be 0,2 mm. With tighter tolerances and maybe better filtration for the pilot flow an even smaller opening could be used.

The actuator of the pilot valve is required to keep the lower orifice closed when the main valve is closed and it is also required to be able to open the upper orifice. Therefore the pressure force pushing the sealing ball upwards cannot be larger than the downward force generated by the actuator. If the force produced by the actuator is assumed to be 5 N bidirectionally, we can calculate with equation (3.3) that the maximum diameter of either orifice in the pilot stage is about 0,8 mm with a 10 MPa pilot pressure level. Because the stem of the solenoid moves through the upper orifice in the pilot valve the upper orifice has to be larger than the lower orifice to have the same flow rate with the same pressure. It is also good to make the lower orifice smaller than the upper one since the return spring generating the closing force in the pilot valve is more extended when closing the lower orifice. Therefore it produces slightly less force because the spring has less tension. A diameter of 0,7 mm will be used for the lower orifice in the following calculations.

When the piston of the cylinder in the pilot stage moves, the piston displaces a volume of fluid which has to flow through the pilot valve. The fluid volume V_p displaced by the movement of the piston can be calculated with equation (3.4) where A_{piston} stands for the area of the piston, D_p for its diameter and l_m for the opening of the main valve i.e. for the movement of the piston.

$$V_p = A_{piston} l_m = \pi (\frac{D_p}{2})^2 l_m \tag{3.4}$$

The pilot valve should be able to provide this fluid volume in the desired response time. The desired response time of the main valve is 2 ms. This can be divided roughly equally to the response time of the solenoid actuator and the time taken for the piston to move. This means that the piston should move from end to end in preferably less than 1 ms. Flow rate Q_p through the pilot valve when the piston is moving can be calculated with equation (3.2) as in chapter 3.1. However, in the case of the pilot valve the pressure difference over the orifice is not simply the pilot pressure since there has to be a certain pressure level in the cylinder's chamber to move the piston.

As determined in chapter 3.1 the cylinder in the pilot stage needs to produce 29 newtons of force in both directions. Earlier in this chapter it was decided that a spring return cylinder should be used for simpler structure of the pilot valve. This means that the 29 N in the other direction has to be produced by the return spring and in the other direction the cylinder has to produce enough force to overcome the spring force and in addition to that it has to produce the 29 N required. So at least 58 N has to be produced by the pressure force in the cylinder. This is the minimum requirement for overcoming the pressure force acting on the poppet of the main valve and therefore for opening and closing the main valve with the maximum operating pressure. The force produced by the piston can be calculated with equation (3.3). The larger the piston is the more force it produces with the selected pilot pressure. However, according to equation (3.4) the larger the piston is the more fluid it displaces. Therefore it also takes longer for it to travel from end to end since the piston's velocity is limited by the maximum flow rate through the pilot valve. This is why the area of the piston has to be carefully optimized.

Let's assume a case where the force of the return spring is the minimum i.e. 29 N and a pressure force of 29 N is also pushing the poppet closed. We can now calculate the pressure required in the cylinder chamber to move the piston with equation (3.3). Then we can calculate the pressure P_p over the pilot valve by reducing the required pressure in the chamber from the 10 MPa pilot pressure. By substituting this difference to equation (3.2) we can now calculate the flow rate through the pilot valve during an opening cycle with equation (3.5) where l_p stands for the opening of the pilot valve. From this flow rate we can calculate the time t it takes for the piston to move i.e. the opening time of the main valve with equation (3.6) where V_p stands for the displaced fluid volume and l_m for the opening of the main valve.

$$Q_p = \mu D_o l_p \sin\alpha \sqrt{\frac{2P_p}{\rho}} \tag{3.5}$$

$$t = \frac{V_p}{Q_p} = \frac{l_m A_{piston}}{\mu D_o l_p \sin\alpha \sqrt{\frac{2P_p}{\rho}}} \tag{3.6}$$

With D_o = 0,7 mm, l_m = 0,4 mm, l_p = 0,2 mm and a piston diameter of 4 mm the equation (3.6) gives an opening response time of 0,77 ms for the piston in the worst case with the 58 N resisting force. We have to remember that only the pressure loss in the pilot valve's orifice is taken to account, not losses anywhere else in the channels or the mechanical friction in the cylinder, flow forces at the main valve poppet etc. Also the kinematics of the valve i.e. the inertia of the piston or the fluid is not taken into account. Flow forces at the poppet are however considered to have a minimal effect on the performance of the valve. Also all the flow channels in the valve should be designed so that they restrict the flow minimally compared to the orifice in the valve.

The effects of the inertia of the piston can be approximated with equations (3.7) and (3.8) where F_p is the pressure force acting on the piston, m is the mass of the piston and l_m is the opening opening of the main valve. The equations describe how fast an object travels a certain distance when in the beginning it has no velocity and a certain net force is subjected to it. If the piston is assumed to have a mass of 3 grams and a constant net force of 20 newtons is assumed to accelerate it, we can calculate that it travels a distance of 0,5 mm in about 0,5 milliseconds. Therefore the inertia of the piston is also an important factor in the response time. The effects of inertia and losses in the hydraulics are both taken into account in the simulations presented later in chapter 4.2.

$$F = ma \tag{3.7}$$

$$l_m = \frac{1}{2}at^2 \rightarrow t = \sqrt{\frac{2l_m}{a}} = \sqrt{\frac{m2l_m}{F}} \tag{3.8}$$

3.3 Solenoid

3.3.1 Basics of solenoid magnetics

A solenoid is an electromagnetic actuator consisting of a coil, an armature which is the moving part in the solenoid, and the body. When current is passed through the coil it creates a magnetomotive force which generates magnetic flux to the solenoid according to Ampere's law. The body and the armature of the solenoid create a magnetic circuit which conducts magnetic flux better than the area outside the circuit. Magnetic flux creates a magnetic force in the solenoid. A cutout of a simple solenoid model is displayed in figure 3.4 and the most important parts are annotated in it.

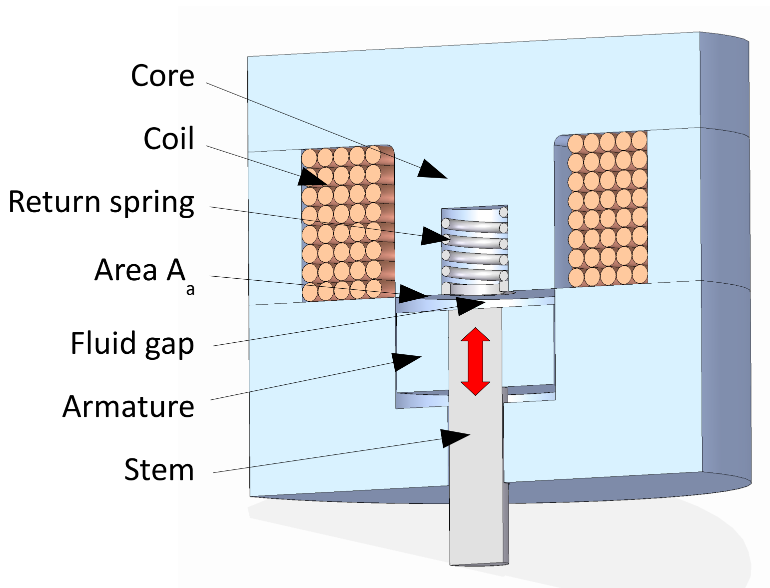


Figure 3.4: A cutout of a simplified solenoid model.

The magnetomotive force can be considered analogous to voltage, magnetic flux analogous to current and the magnetic circuit analogous to electric circuit. The magnetomotive force U_m generated by the coil depends on the current passing through the coil and the number of wire loops in it. The magnetomotive force can be calculated with equation (3.9) where N is the number of loops in the coil and I is the current passed through all of the loops. [17, p. 205]

$$U_m = NI \tag{3.9}$$

Magnetic reluctance measures the magnetic circuit's ability to resist magnetic flux created by the magnetomotive force. Reluctance can be considered analogous to electrical resistance. The magnetic reluctance of a certain segment of the magnetic circuit depends on the permeability of the magnetic circuit's material, cross sectional area of the part and length of the segment. Permeability is a material property which is explained in more detail in chapter 3.3.2. Magnetic reluctance of the whole magnetic circuit can be approximated segment by segment with equation (3.10) where l_s is the length of the segment, A_c is the cross sectional area of the segment, μ_0 the permeability of vacuum and μ_r is the relative permeability of the material. [17, p. 206]

$$R_m = \frac{l_s}{\mu_0 \mu_r A_c} \tag{3.10}$$

The total reluctance of the magnetic circuit is the sum of all of its component's reluctance. Because the magnetic circuit in a solenoid is usually almost totally made of high permeability, and therefore low reluctance, steel the fluid or air gap between the solenoid's core and the armature is the largest factor contributing to the total reluctance of the magnetic circuit. In the design presented in this thesis when the solenoid is open the gap creates about 90 % of the total reluctance. The reluctance of the rest of the magnetic circuit is mostly important when the solenoid is closed i.e. the pilot valve is held open. In this situation in ideal case the distance between the core and the armature is zero, and therefore the reluctance of the fluid gap is also zero. Then the reluctance of the rest of the magnetic circuit has a significant effect on the amount of current which is required to hold the valve open.

If the magnetic flux is assumed to be evenly distributed in the material, the flux density B in any part of the circuit can then be estimated with equation (3.11) where R_{tot} is the total reluctance of the magnetic circuit and A_c is the cross sectional area of the examined flux path. [2, ch. 3.1]

$$B = \frac{\frac{U_m}{R_{tot}}}{A_c} \tag{3.11}$$

When the magnetic flux is assumed to be evenly distributed between the core and the armature of the solenoid and no magnetic flux is assumed to flow through the bottom of the armature, the pulling force F_m generated by the solenoid depends on the magnetic flux density and the area A_a between the solenoid's core and the armature according to equation (3.12). [2, ch. 5.3]

$$F_m = \frac{B^2 A_a}{2\mu_0} \tag{3.12}$$

Equation (3.12) shows that the force generated by the solenoid grows exponentially as the magnetic flux density through the top of the armature increases. However, the magnetic flux density is limited by the saturation of the material of the magnetic circuit. The saturation phenomenon is explained in detail in chapter 3.3.2. It is why the solenoid should be designed so that the magnetic flux density in the material is never expected to rise over the saturation magnetic flux density but instead the maximum required force is generated with a magnetic flux density a little under saturation. Also the size of the armature should be kept as small as possible because of inertia and space limits. This leads to conclusion that when the magnetic flux density in the solenoid is limited by the saturation, the solenoid's maximum force is limited mainly by the area A_a between the armature and the core. The surface area of the armature should also be about the same as the cross sectional area of the core since the magnetic flux density in both of them has to be under the saturation limit.

By assuming that all of the reluctance in the magnetic circuit comes from the fluid gap, with length l_p, between the core and the armature and substituting equations (3.10) and (3.11) to (3.12) we get equation (3.13).

$$F_m = \frac{U_m^2 A_a \mu_0 \mu_r^2}{2 l_p^2} \tag{3.13}$$

From equation (3.13) can be seen that with a certain magnetomotive force U_m as input, increasing the length of the fluid gap l_p has an exponential effect in decreasing the generated force. This is why the travel distance of the armature i.e. the opening of the pilot valve should be kept at the absolute minimum where the hydraulic pilot stage can still work and produce enough flow. The desired travel distance of the armature was decided to be fixed to 0,2 mm for reasons explained in detail in chapter 3.2. From equation (3.13) could also be concluded that increasing magnetomotive force or relative permeability also increases the produced force exponentially but in real life the saturation limit of the magnetic circuit material prevents it. Therefore the saturation magnetic flux density and the area between the armature and the core are the most important parameters in determining the force generated by the solenoid in a stationary state.

The designing of the actuator, the pilot stage and the main valve are related so that the force generated by the actuator in the pilot stage needs to be at least the same as the pressure force acting on the sealing element of the pilot valve. The pressure force depends on the pilot pressure and the size of the orifices in of the pilot valve. The pilot pressure and the orifice sizes affect the response time of the main valve. So on the other hand a larger actuator for the pilot stage would enable controlling larger flow to the piston controlling the main stage. This would result to a faster response of the main valve. On the other hand the valve is required to be miniaturized so a compromise is needed. The selected compromise is that the pressure force on the sealing element of the pilot valve should not exceed 5 N. Therefore the solenoid should produce 5 newtons of closing force and the necessary forces to accelerate the armature during its opening and closing movements. Since the solenoid itself only produces unidirectional force there needs to be a spring to produce the force in the other direction. Thus the solenoid needs to produce at least 10 newtons of force to accelerate the armature in addition to overcoming the force of the return spring.

3.3.2 Material selection

An important part of solenoid design is choosing the material of the magnetic circuit. The three most important material parameters are permeability, saturation magnetic flux density and electrical conductivity.

Permeability describes the material's ability to support an external magnetic field. Permeability of a material is usually expressed as a relative permeability which is a multiplier of permeability of the vacuum. Materials can be divided to three main groups according to their permeability. The most important group for magnetic applications is ferromagnetic materials. Ferromagnetic materials have a relative permeability of significantly more than one which means that they are significantly magnetized in an external magnetic field. [3, p. 196]

Ferromagnetic materials cover mainly iron, cobalt, nickel and their alloys. They consist of small magnetic domains which normally are pointing at arbitrary directions. Thus in a larger scale the material does not have a net magnetic polarity. In external magnetic field these small magnetic domains line according to the external field lines magnetizing the material itself, which strengthens the external magnetic field. Most other materials, besides ferromagnetic materials, have a relative permeability of about one. Permeability is somewhat analogous to electrical conductance. Therefore a large magnetic flux can be created with a lower input power to a magnetic circuit with high permeability. [3, p. 196]

When a ferromagnetic material is in an external magnetic field the small magnetic domains start to line according to the magnetic field lines. The stronger the magnetic field is the better the domains are aligned and the more the material is magnetized. A property called coercivity measures a material's ability to resist magnetization. When all the domains are perfectly aligned according to the magnetic field lines, the material cannot be magnetized further. After that the material is said to be saturated and it can no longer support the increase of magnetic flux density and therefore its permeability falls to the same as the permeability of vacuum. After saturation magnetic flux density, the magnetic flux density in the material can still be increased but it requires a very large magnetic field and is often not practical. [3, p. 197]

The saturation magnetic flux density is a property dependent on the composition of the material and its annealing state. Figure 3.5 shows an approximation of the magnetic flux density of AISI 12L14 low carbon steel as a function of the magnetic field in the material. Hysteresis caused by the coercive force is neglected in the curve. From the figure can be seen that the saturation magnetic flux density of AISI 12L14 is about 1,8 teslas. It can also be seen that in the beginning of the magnetization, magnetic flux density rises very fast but the rise slows down as the magnetic domains start to be better and better aligned. This means that when AISI 12L14 is magnetized only a little, its permeability is higher than when the magnetic flux density is getting closer to the saturation limit. With equation (3.12) can be calculated that an area of about 8 mm^2 is required between the core and the armature to produce 10 N of force when the material is saturated with magnetic flux. In practice the area should be larger to increase the force and thus to reduce the response time of the solenoid.

Because the response time of the solenoid is expected to be in the order of milliseconds, the dynamic phenomena become important also in

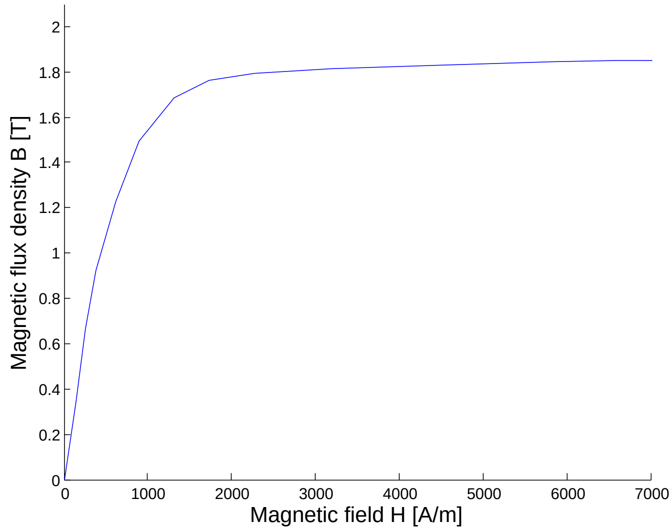


Figure 3.5: The magnetization curve of AISI 12L14 low carbon steel.

the magnetic circuit. The most important of these is eddy currents, which are very much affected by the conductivity of the magnetic circuit's material. When magnetic flux is generated by the current in the coil, the increasing flux also generates secondary current loops into the core material while passing through it. These loops absorb energy from the flux at the same time slowing its increase. Eddy currents also prevent the magnetic flux from penetrating inside the core immediately, causing an effect called skin effect. It makes the magnetic flux concentrate on the surface of the magnetic circuit in transient situations. This is an unwanted phenomenon since we would like the magnetic flux to rise as fast as possible in the magnetic circuit to get a fast response from the solenoid. Unfortunately eddy currents are always induced into electrically conductive material when the magnetic flux in it is changing and the only thing that can be done is to reduce the material's conductivity. With reduced conductivity the eddy current loops conduct less current and the energy loss from the flux is reduced. [4, ch. 12.2]

Reducing the electric conductivity of the magnetic circuit can be done in at least three ways. The easiest and the most common way is to make the circuit from an alloy with a suitable composition. Iron, which is the most used magnetic material, can be alloyed with for example aluminum or silicon to decrease its conductivity significantly. Unfortunately these alloys also usually have a worse permeability and saturation magnetic flux density than pure iron.

The second option is to glue together thin sheets of metal as a stack to create a solid block with a laminated structure. Because the glue is not electrically conductive, the block has an anisotropic conductivity. In the plane of the layers the conductivity is the same as the base material's conductivity, but orthogonally to layers the conductivity is very low. The problem is that if the magnetic flux is passing through the material orthogonally to the layers, eddy currents form inside each layer as they would do in a homogenous material. This is why the magnetic circuit would have to be designed so that magnetic flux passes mainly in the direction of the layers. This can however be difficult since the layered material is not very strong and it is difficult to machine complex structures from it. [4, ch. 13.2]

The third option are composites based on combining metal powder and insulating material. The composites consist of fine iron powder particles, about 50 to 100 μm in diameter, and each particle is coated with an insulating layer. This powder is compacted and cured to form a solid block. Since each particle is electrically insulated from the others the conductivity of the cured material is very low which eliminates eddy currents. On the other hand the insulating layer also decreases the permeability of the material to the order of a few hundred and also significantly reduces the saturation magnetic flux density and thus the maximum force of the solenoid. [4, p. 468]

In conclusion a composite material would be the best option for achieving the lowest response times but it would also lead to higher coil currents and probably even a larger armature. The manufacturing process of soft magnetic composites is also not very suitable for prototyping since it requires an expensive mold. The mechanical structure of the designed valve requires quite extensive machining of the core material and therefore a layered material was also not a good option. Thus a suitable homogenous alloy was needed. AISI 12L14 low carbon steel was determined to be suitable since it has a quite high permeability and saturation magnetic flux density but also its electrical conductance is significantly lower than the conductivity of pure iron.

3.3.3 Coil

Also the properties of the coil are an important factor in solenoid design. The coil is actually the largest part in the actuator and the limiting factor in many ways. The most important properties of a coil, in addition to its physical size, are its resistance and inductance.

Resistance R of the coil affects the ohmic losses in the coil. When current is passed through the coil, heat is generated in the coil with power P

which can be calculated with equation (3.14) [27, p. 110].

$$P = RI^2 \tag{3.14}$$

Resistance of a copper wire can be calculated with equation (3.15) [27, p. 110] where ρ_c is the resistivity of copper, l_w is the length of the wire and A_w is the cross sectional area of the wire.

$$R = \frac{\rho_c l_w}{A_w} \tag{3.15}$$

The length of the wire in the coil can be approximated with the number of loops and the mean diameter of a loop in the coil. The approximation is presented in equation (3.16) where D_i is the inner diameter of the coil and w_c is the thickness of the coil. The dimensions are illustrated in figure 3.6.

$$l_w = N\pi(D_i + w_c) \tag{3.16}$$

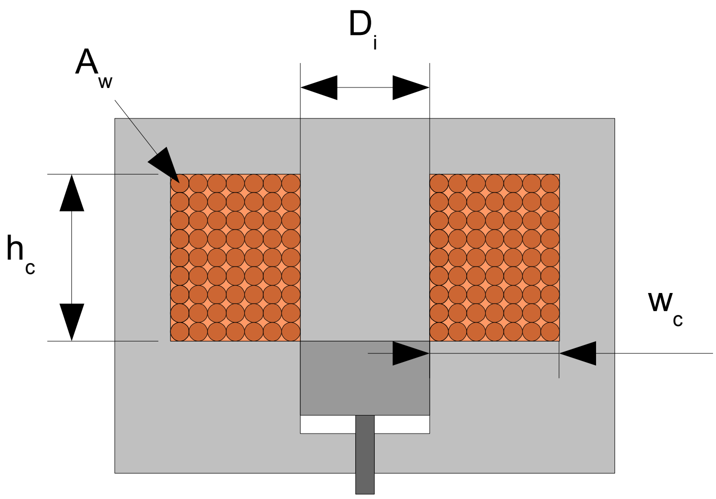


Figure 3.6: The dimensions of the coil.

Maximum cross sectional area of the wire with a certain number of loops and limited space can be approximated with equation (3.17) where h_c is the height of the coil and f is a fill factor describing the amount of cross section filled by copper. A 50 % fill factor is a conservative approximation when considering that the wire is round and the insulation lacquer on the wire requires space. [27, p. 111-113]

$$A_w = \frac{h_c w_c f}{N} \tag{3.17}$$

Now the resistance of the coil can be approximated with equation (3.18) where equations (3.16) and (3.17) have been substituted to equation (3.15).

$$R = \frac{\rho_c N^2 \pi (D_i + w_c)}{h_c w_c f} \tag{3.18}$$

Inductance L describes the coil's ability to resist the change of the current in the coil which then limits the response time of the solenoid. The inductance of the coil is determined by the reluctance R_{tot} of the magnetic circuit surrounding it, and the number of loops in the coil according to equation (3.19) [2, p. 74].

$$L = \frac{N^2}{R_{tot}} \tag{3.19}$$

Because the reluctance of the magnetic circuit in the solenoid depends on the position of the armature, also the inductance is dependent on it. In our case we are mostly interested in the situation where the armature is the furthest away from the core, for example in the beginning of a valve opening cycle. This is usually the situation where the current should rise fastest. Ideally the current in the coil could be changed stepwise when the solenoid is actuated. This would provide instantaneous magnetomotive force for the magnetic circuit. Instead the current starts to rise asymptotically to the maximum value according to equation (3.20) [2, p. 234]. In equation (3.20) I_{max} is the maximum current determined by the supply voltage U and the resistance of the coil and τ is the time constant of the RL-circuit. The time constant is determined by the resistance and the inductance of the coil.

$$I(t) = I_{max}(1 - e^{\frac{-t}{\tau}}) = \frac{U}{R}(1 - e^{\frac{-tR}{L}}) \tag{3.20}$$

In the designed solenoid the total reluctance of the magnetic circuit is about 8 MH when the valve is closed. The resistance of the coil with 80 loops of 0,35 mm^2 wire is about 0,45 Ω. Therefore it can be calculated with equation (3.19) that the inductance of the coil is 0,8 mH. With equation (3.20) can then be calculated that with a 24 V supply voltage it takes about

0,25 ms for the current to rise to 7 amperes in in the coil. Then the magnetomotive force produced by the coil is 560 A. The same magnetomotive force can be produced with for example 160 loops in the coil and a 3,5 A current. However, then the resistance of the coil would be 1,8 Ω and the inductance 3,2 mH. This leads to about 0,55 ms rise time to 560 A magnetomotive force.

When a certain magnetomotive force is required from the coil, the current required to generate it depends on the number of loops in the coil according to equation (3.9). The heating power of the coil while generating the magnetomotive force can be calculated by substituting equations (3.9) and (3.18) to equation (3.14).

$$P = RI^2 = \frac{\rho_c N^2 \pi (D_i + w_c)}{h_c w_c f} (\frac{U_m}{N})^2 = \frac{\rho_c \pi (D_i + w_c)}{h_c w_c f} U_m^2 \tag{3.21}$$

As can be seen from the equation (3.21) the heating power of the coil when creating a certain magnetomotive force is not actually dependent on the number of loops or the length of the wire in the coil but mostly on the physical dimensions of the coil. This means that the number of loops can be optimized for fast current rise without affecting the heating. An ideal situation regarding response time would be a coil with only one loop but in that case the required current would be very high.

Heating creates unnecessary power losses in the coil but it also limits the downsizing of the coil. If the coil is made too small its resistance gets too large which increases ohmic heating. A small coil also conducts less heat to the surroundings leading to an even higher temperature. Heating can be a problem if the holding current heats the coil to too high temperature, or if in dynamic loading the boosting pulses generate too much heat. A too high temperature may melt the insulating lacquer on the coil's wire.

Because of the problems caused by the inductance and the resistance of the coil, a method called boosting is used when good performance is required from a solenoid. Boosting means that when the current in the coil has to be changed fast, for example when opening a valve, a large voltage is applied to the coil so the current rises fast according to equation (3.20). The voltage cannot be kept large for long periods because the large current caused by it creates too much heat. Therefore the boost voltage is applied only for a few milliseconds until the valve is opened. After that only a small voltage is applied to the coil. This voltage is just enough to create enough current and thus enough magnetomotive force to keep the valve open.

Chapter 4

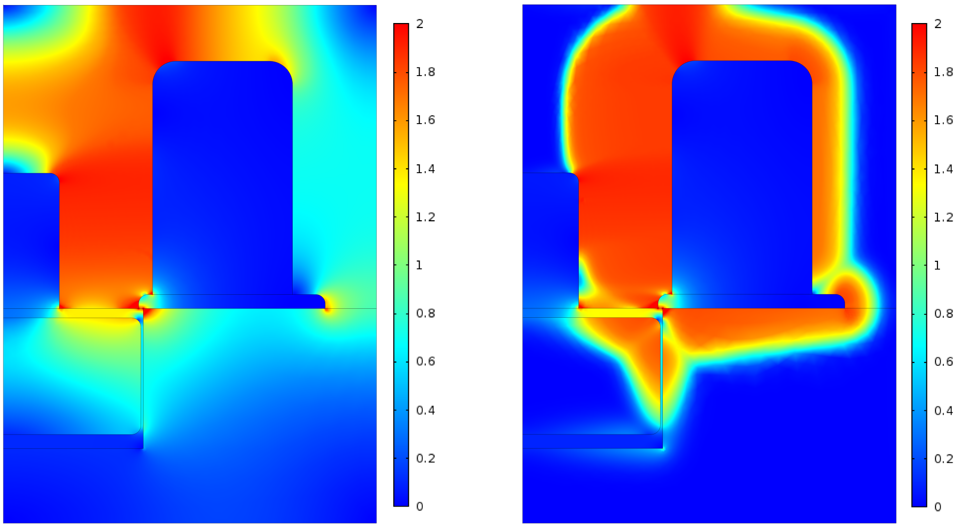
Simulations

4.1 FEM simulation of the magnetic circuit

The first approximations of the solenoid's dimensions were calculated by iterating the equations presented in chapter 3. After that the magnetic circuit of the solenoid was also simulated with Finite Element Method (FEM) to get better approximations of the generated force and to see the effects of saturation and eddy currents. The programs used for FEM calculations were Comsol Multiphysics versions 3.5 and 4.2 and Finite Element Method Magnetics (FEMM) version 4.2. The nonlinear BH curve of AISI 12L14 steel presented in chapter 3.3.2 was input into both programs to take into account also the saturation in the simulations. In both programs a 2D axial symmetric model was used for the solenoid.

In Comsol Multiphysics simulations were made both in stationary and transient modes. The stationary mode calculates the distribution of the magnetic field in a stationary state which forms in time with a constant magnetomotive force. This is mainly useful for calculating the required magnetomotive force to keep the valve open and to approximate the level of opening force. The force generated by the solenoid can be calculated from the distribution of the magnetic flux or the Maxwell stress tensor at the boundaries of the armature [2, ch. 14.4]. The force can also be calculated with analytical equations as in chapter 3 but it is easier to see the effects of saturation in a FEM simulation. A plot of the magnetic flux density in a stationary simulation with a 320 A magnetomotive force from the coil is displayed on the left side of figure 4.1.

The transient mode in Comsol Multiphysics calculates the changing magnetic field in the magnetic circuit at certain time steps for a selected time period. The results show how the magnetic field and the flux density



(a) Stationary simulation with a 320 A magnetimotive force

(b) Transient simulation at t = 0,7 ms

Figure 4.1: Plots of the magnetic flux density [T] in stationary and transient simulations in Comsol Multiphysics.

develop in the solenoid in time after the voltage is switched to the coil. The current was approximated with either equation (3.20) or other similar smooth curves. Transient simulations clearly show the skin effect caused by the eddy currents. From the results of the simulations can also be calculated how fast the solenoid can generate the required force. A plot of the magnetic flux density 0,7 ms after the beginning of a transient simulation is displayed on the right side of the figure 4.1. In the figure can be seen that the magnetic flux is concentrated in the core and the flux density is smaller in the outer parts of the magnetic circuit. Also some fringing of the magnetic flux is visible around the fluid gap between the core and the armature.

There were problems in getting the transient simulations in Comsol Multiphysics to converge and therefore several corners in the models are rounded even though in real life they are sharp. The elimination of sharp corners from the model reduces the number of singularities in the calculations and thus lets the simulation converge more easily. Because of the problems with the convergence, the current levels used as the excitation for the magnetic circuit in the simulations had to be kept quite low. Therefore the response time results from the simulations made with Comsol Multiphysics can be conservative.

Also FEMM was used in stationary mode and the results of the simulations were compared with simulations made in Comsol Multiphysics. The results seemed to be very similar. FEMM cannot compute the transient response of the magnetic circuit but instead it calculates a time harmonic analysis of the problem. This means that the input to the model of the magnetic circuit is a sinusoidal current and the results of the simulation are the fundamental components of the resulting magnetic field and flux density.

Strictly speaking time harmonic analysis applies only to linear problems. Because of the magnetic saturation, the permeability of the material is nonlinear which makes the whole problem nonlinear. This also has a significant effect on the results. FEMM however tries to take the nonlinearity of the problem into account by assuming that the magnetic field resulting from the current is sinusoidal and then adjusting the resulting magnetic flux to better follow the nonlinear BH curve defined by the user. This method results to larger magnetic flux densities than in real life so the results from FEMM could not be used for calculating the force generated by the solenoid when the material was saturated. They however were useful in comparing different geometries and material combinations since the time harmonic analysis is much faster to calculate than the transient response and it does give some information of the speed of the transient response. [21]

When the magnetomotive force is created by a sinusoidal current the amount of magnetic flux in the magnetic circuit is fairly well described by the fundamental component as long as the material is not severely saturated. The fundamental component is affected by the electrical conductivity and the permeability in the same way as the results of the transient simulations. Therefore the response time of the magnetic circuit can be approximated from the amplitude of the fundamental component calculated with time harmonic analysis. An image of the magnetic flux density in the magnetic circuit calculated with time harmonic analysis in FEMM is the displayed in figure 4.2. The image shows clearly how the magnetic flux is concentrated on the surface of the magnetic circuit because of the skin effect. However in the image it is visible that there are magnetic flux densities of over 2,3 T even though the saturation magnetic flux density of the material is 1,8 T.

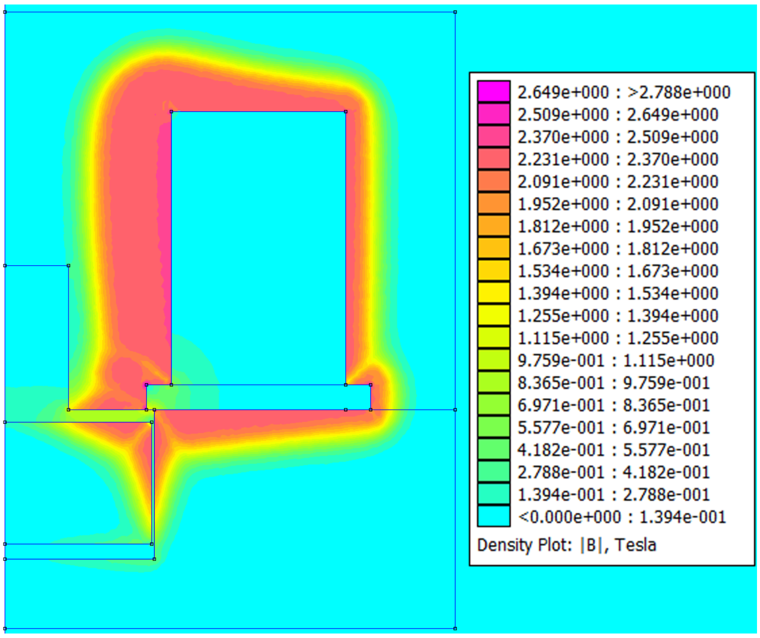


Figure 4.2: Plot of magnetic flux density from a 500 Hz time harmonic analysis in FEMM.

4.2 Simulink-model of the valve

A simulation model of the whole valve consisting of models for the kinematics of the solenoid actuator and the pilot stage and flows of the main and pilot valves was made in Matlab Simulink. The model simulates a single opening cycle of the valve. Input for the simulation is the result of a transient simulation in Comsol Multiphysics describing how the magnetic flux density at the upper surface of the armature develops with respect to time. The model also uses variables from the same scripts that were used to calculate the dimensions in chapter 3. This makes it easy to simulate different geometries and their combinations and to get an approximation of their performance. Block diagram of the Simulink model is displayed in figure 4.3.

The model first calculates the force generated by the solenoid from the magnetic flux density according to equation (3.12). Then the movement of the solenoid's armature is calculated with equations (3.7) and (3.8) which describe the kinematics. The model also takes into account the pressure force resisting the armature's movement in the fluid volume and flows through the armature according to equations (3.3) and (5.1). The armature's position, which is calculated in the leftmost block of figure 4.3, is fed into the next block which simulates the pilot stage of the valve. This block calculates the flows through the two orifices in the pilot valve with equation (3.2). The pressure increase in the fluid volume under the main valve piston is calculated from the integrated flow through the orifices.

In the beginning of the simulation the pressure in the pilot channel is zero but when the lower orifice of the pilot valve is opened the pressure starts to rise. The model also takes to account the compressibility of the fluid. The calculated pilot pressure is fed to the third block. The third block simulates the main valve's poppet with kinematic equations. It takes into account the force from the return spring and pressure force on the poppet. When the pilot pressure has risen enough so that the pressure force can overcome the force from the return spring of the main valve, the main valve piston starts to move. The position of the piston is fed back to the pilot valve block so that the changing fluid volume in the pilot channel can be taken to account in the pressure calculations.

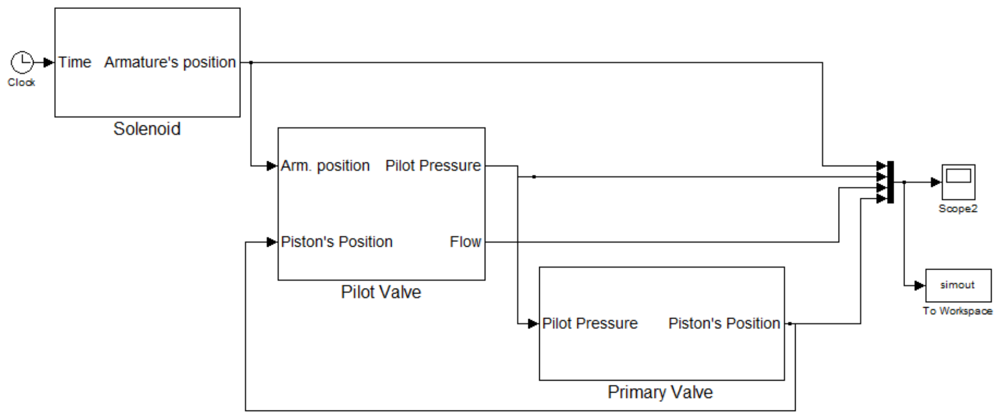


Figure 4.3: The block diagram of the Simulink-model of the whole valve.

The most important input for the model is the transient magnetic flux for a time period of 2 ms simulated with Comsol Multiphysics. The flux is simulated with a static geometry meaning that during the simulation the solenoid's armature stays at the same position 0,2 mm away from the core. However, in real life and in the Simulink-model the armature starts to move as soon as the magnetic force overcomes the force of the return spring. When the armature starts moving, the reluctance between the armature and the core decreases, which has an effect on the magnetic flux.

The simulation made in Comsol Multiphysics does not take into account this effect and in this matter the simulation of the magnetic flux is not very realistic.

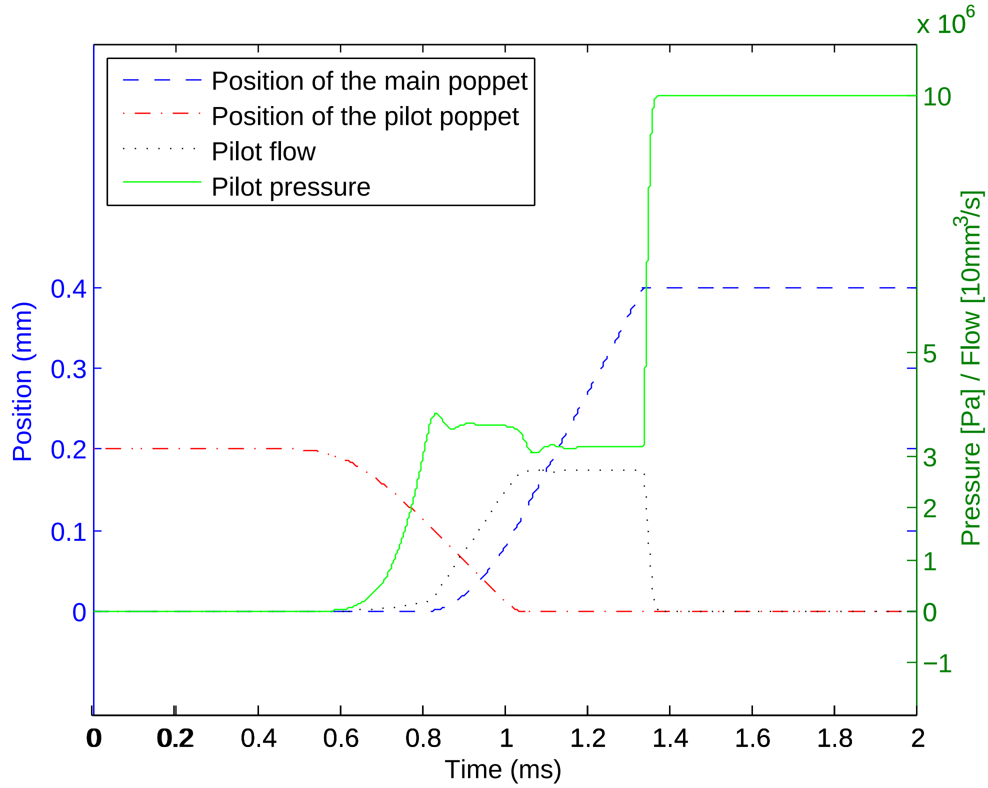


Figure 4.4: The simulated positions of the pilot and the main poppets, the flow through the pilot valve and the pilot pressure during an opening cycle of the valve.

The results of the Simulink-simulation are presented in figure 4.4. The magnetic flux density starts increasing at the beginning of the simulation just as it would do on real life if voltage to the coil would be switched on at t = 0. The simulation predicts that the armature starts to move at about t = 0,6 ms. The pilot pressure starts to rise at about the same time. The pilot pressure rises to about 3,5 MPa and the main piston starts to move at about t = 0,8 ms. The pilot pressure stays at the same level while the main piston is moving because the force resisting the movement i.e. the force of the return spring is constant in the simulation. The flow to the pilot channel is limited by the small orifice in the pilot valve and therefore a balance between the force of the return spring and the pressure force on the piston forms while the piston is moving. After the piston has reached its destination, in this simulation 0,4 mm opening, the pressure in the pilot channel reaches quickly the pressure of the pilot pressure supply line. According to the simulation the main piston is at its destination and the main valve is fully open at about 1,4 ms.

Chapter 5

Prototyping

5.1 Actuator prototype

A prototype of the solenoid actuator for the pilot stage was built to validate the FEM simulations of the magnetic circuit and the force calculations based on FEM and analytical equations. The prototype was designed to be as easy to manufacture as possible but so that it still retains the important features that the final design would contain. A CAD model of the designed prototype is displayed in figure 5.1.

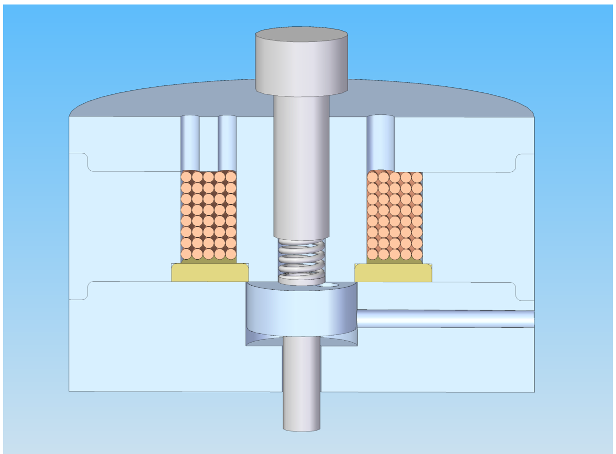


Figure 5.1: A cutout of the actuator prototype's CAD model.

The prototype consists of three turned parts which are attached together with three bolts to create the body of the solenoid. The parts form two cavities between them. The upper cavity houses the coil and the lower houses the armature. All three parts as well as the armature, and therefore the whole magnetic circuit are made of AISI 12L14 low carbon steel. All these parts are axially symmetric since it makes the manufacturing easier. The armature moves vertically inside the bore in the lowest part of the body. In the center of the armature there is a hole for an interference fit of the stem which actuates the pilot valve. In the actuator prototype the stem is however used only for measuring the position of the armature. There should be as small as possible a gap between the sides of the armature and the bore where it moves in since the smaller the gap is the less reluctance is causes to the magnetic circuit. There is also a plastic seal around the bottom part of the core. It was designed to prevent fluid from leaking from around the armature to the cavity housing the coil and from there through the wire inlets to outside the prototype.

A round armature with a hole in the center would be easy to manufacture by turning, but there is one complication. Normally in a valve structure the bore would fill up with fluid leaking from the flow channels. When the armature moves in the bore it needs to displace the fluid from its way. Both the top and the bottom parts of the bore are basically sealed volumes so there needs to be a flow channel for the displaced fluid from the top volume to the bottom volume. In the actuator prototype the flow channels were made by drilling two vertical bores through the armature.

The required flow rate through the armature is dependent on the diameter of the bore it moves in and the speed of the armature. When the armature moves there needs to be a pressure difference between the upper and the lower volume to create the necessary flow between them. The pressure difference Δp can be calculated with equation (5.1) where Q is the flow rate through one of the orifices, η is the dynamic viscosity of the fluid, l_c is the length of the channel and d is the diameter of the channel [12]. When applying this equation, the flow channels through the armature are considered circular pipes and the flow in them is considered laminar. This pressure difference across the armature also causes a force which slows down the movement of the armature. The diameter of the flow channels was selected to be 1,3 mm to make the resisting force small enough so that the slowing down is not very significant. This force is taken into account in the Simulink-model discussed in chapter 4.2.

$$\Delta p = \frac{Q128\eta l_c}{\pi d^4} \tag{5.1}$$

Because the armature's stem moves in and out from the prototype when the armature is moving, the fluid volume around the armature also changes. The reduction in the fluid volume around the armature, when the armature moves upwards, forces some of the fluid out through the bore where the stem moves in. Therefore, in the tests additional fluid had to be injected into the bore continuously. This is why there is a small channel in the lower part of the body horizontally from the side of the armature to outside of the prototype.

A return spring was also needed for the prototype. The minimum spring force required to overcome the pressure force in the pilot stage was determined to be about 5 N in chapter 3.3. The force of the return spring has to be larger than that to ensure that the pilot valve closes reliably and fast. The spring is placed between the core and the armature into a bore drilled into the core. Therefore it reduces the area where the magnetic flux passes between the core and the armature. This is why the spring should have as small as possible a diameter so it would not reduce the area significantly. The chosen spring has a maximum force of 8,59 N, a diameter of 2,4 mm and a free length of 3,86 mm. It was the smallest spring found which could produce a large enough force. The pretension and therefore the downward force of the return spring can be adjusted with a screw. By iterating the equations presented in chapter 3 and refining the results with FEM simulations the prototype was decided to have the dimensions presented in figure 5.2.

The machined parts for the prototype are displayed in picture 5.3. The parts were machined by the author mainly with a lathe. The most difficult part to machine was the armature which is displayed in picture 5.3 second from the left. In the picture also the stem is attached to the armature. The whole part is only 6 mm in diameter and 2,5 mm long but in addition an attachment hole for the stem has to be drilled in the center and two additional holes for the flow channels. This can be done manually, but requires some precision and time. For manufacturing a larger batch of armatures an alternative method, described in chapter 5.2, was designed. The coil has 80 loops of 0,35 mm diameter wire and it was wound by hand in a lathe.

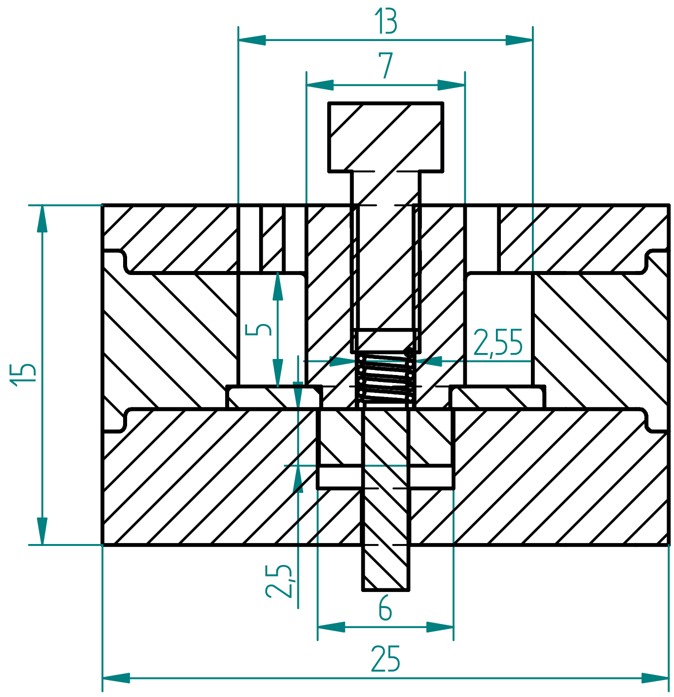


Figure 5.2: The dimensions of the actuator prototype in millimeters.

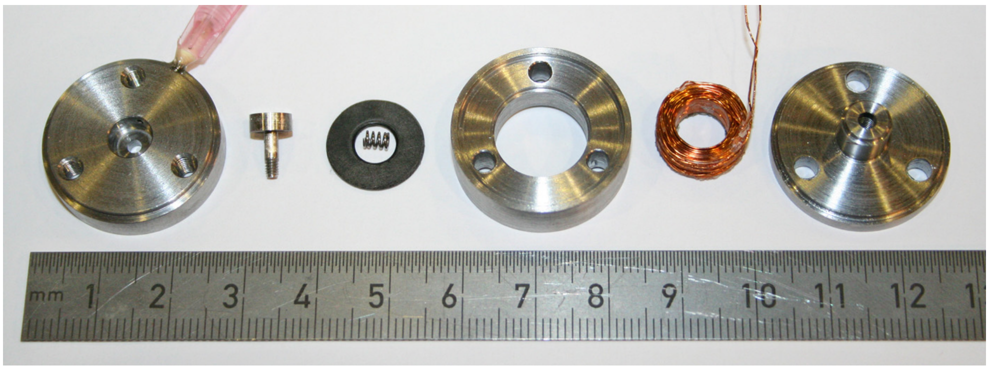


Figure 5.3: The actuator prototype disassembled. From left to right: lower part of the body, armature and its stem, plastic seal with the return spring in the middle, middle part of the body, coil and upper part of the body.

5.2 Valve prototype

5.2.1 Structure

After the actuator prototype was tested and it was found to satisfy the requirements, a valve prototype combining the actuator with the pilot and main valves was required. The testing and properties of the actuator prototype are discussed in chapter 6.1. A prototype with a single valve would be enough to prove that the designed valve works, however it was decided that it would not take much more effort to make a DFCU prototype with four similar valves. A DFCU with four valves could also be applied to some real applications.

The main idea in designing the structure of the valve package was to reduce the number of individual parts to minimum, which would reduce manufacturing and assembly costs in the future. Every on/off valve needs a coil, a return spring and an armature for the solenoid, a stem for the armature, a sealing ball and a return spring for the pilot valve and a piston and a return spring for the main valve. The rest of the valve is flow channels and magnetic circuits. These can be machined to separate layers which are attached together to form the body of the DFCU and the magnetic circuits and flow channels between the layers. The idea is illustrated in figures 5.5 and 5.4.

One problem with this kind of design is that there may be leaks between the layers. Leaking can be either external, meaning that the fluid leaks out from the valve, or internal, meaning that the fluid leaks from one channel to an other inside the valve. The amount of leaking depends on how the layers are attached together. In this prototype the layers are attached with screws for easy disassembling. This however does not make the contact between the layers leak proof even though the layers are ground to a fine surface roughness. When the valves are required to be leak-free internal leaking may be more problematic than external leaking since the distance between the machined flow channels in the layers are very small, only some millimeters. This makes it impossible to place for example an O-ring seal between the channels. On the other hand the channels of the whole DFCU can be surrounded by a large O-ring which prevents external leaking. This prototype has O-rings surrounding all the channels between all the layers. Leaking between the flow channels could be prevented for example by sealing the layers together with sealing glue or the layers could be soldered to each other.

The number of layers needed for the structure should be as low as possible to minimize the amount of leaking and to make the structure easier to

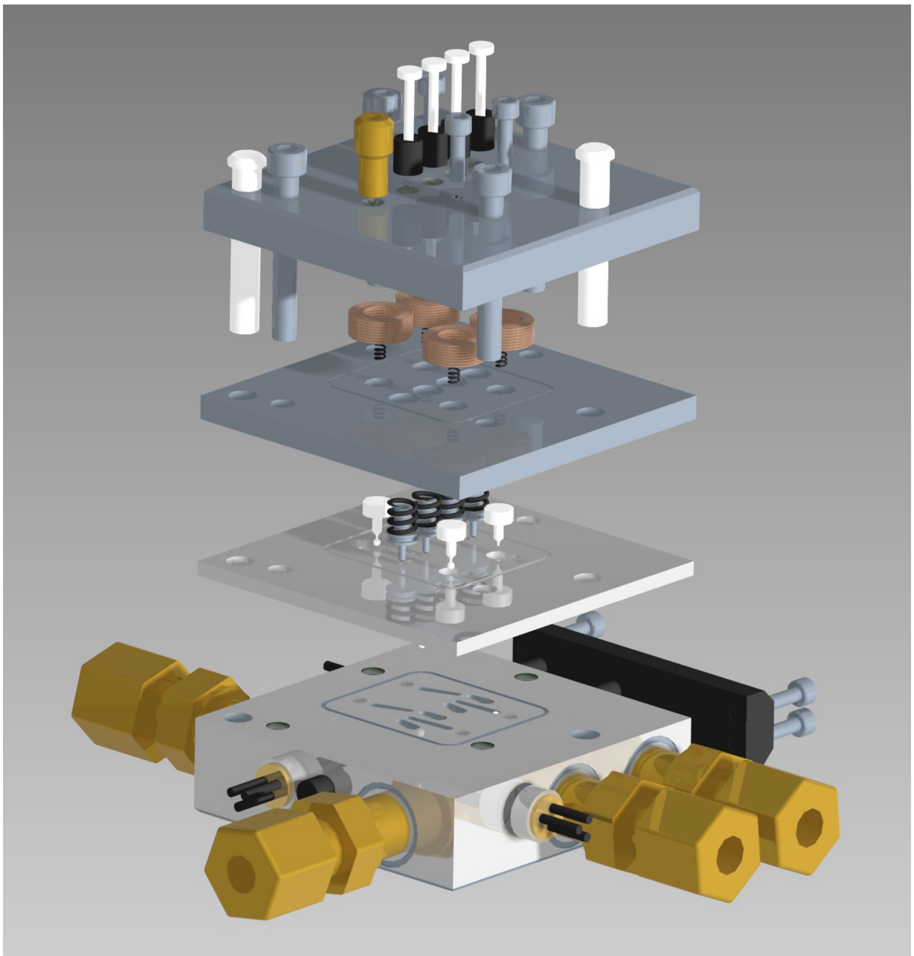


Figure 5.4: An exploded view of the valve prototype's CAD model.

manufacture. The flow channels in a pilot operated valve however need to be somewhat complex which means that several layers are required. Also to make the machining easier, preferably only one side of the layers should be machined to avoid any misalignments of the blank while changing its orientation.

The DFCU prototype in this thesis consists of four separate layers and only the top layer is machined on both sides. An exploded view of the CAD model of the DFCU prototype is displayed in figure 5.4. The two upper layers of the valve package house the solenoid and the main valve's piston. These layers are made of AISI 12L14 steel because of its magnetic properties. Two layers are necessary for housing the coil and the armature completely between them to make a complete magnetic circuit. The

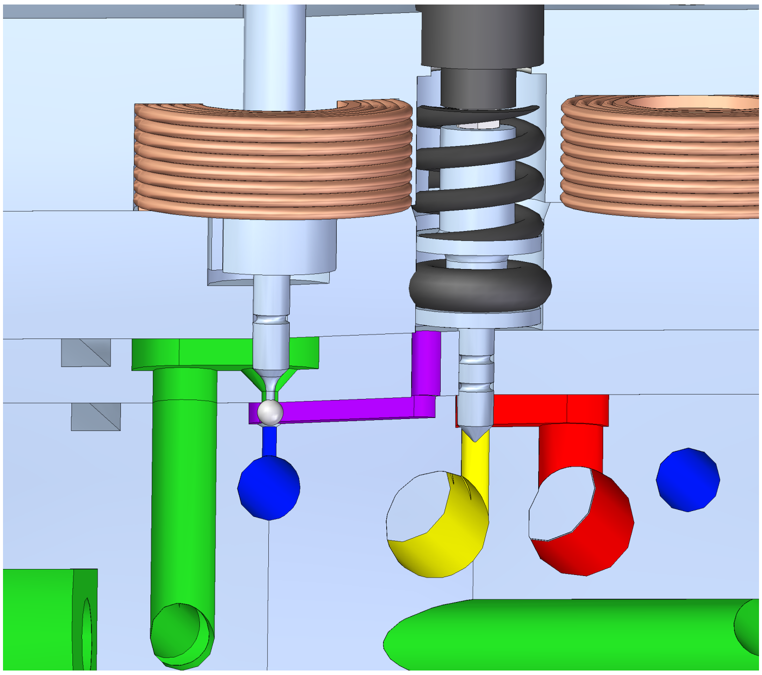


Figure 5.5: A cutout of a CAD model of one on/off valve.

armature should also be separated from the main flow channels so that any force caused by flow cannot affect it, since the magnetic force and the spring force are small compared to the hydraulic forces present in the flow channels. The third layer from the top contains the upper orifice for the pilot valve, milled flow channel from the orifice to the pilot stage's tank channel, drilled flow channel from pilot channel to under the piston and a bore for the poppet of the main valve. The bottom block contains both the lower orifice for the pilot valve and the main valve's orifice, milled flow channels on top of it and drilled flow channels inside the block for pilot and main valve pressure and tank channels. Images of all of the layers are displayed in Appendix A.

The bottom block is also designed so that in a larger valve package the valves could be mounted both on top of it and under it. This way valves on both sides of the block can use the same flow channels so more valves can be fitted to the same volume. It is also more practical to use a large flow channel instead of several small channels since larger channels are easier to drill and the pressure drop across a channel gets exponentially larger when reducing the diameter of the channel.

The flow channels which are drilled into the bottom layer are sealed at the sides of the layer with step screws that block the end of the flow channel. In figure 5.5 the blue flow channel is the pilot pressure supply, the violet channel is the pilot channel and the green channel is the tank channel for the pilot valve flow. Red and yellow channels are the main flow channels.

A disadvantage with pilot operated valves is that there needs to be a connection between the pilot stage and the main stage to transmit the power required to actuate the sealing element of the main valve. In this prototype the connection is the stem of the piston which moves in a bore which is between the pilot channel and the main flow channel. This connection is difficult to seal completely and therefore it causes some leakage between the pilot and the main channel. This leaking causes some power loss which may or may not be important depending on the application.

If the leaking affects the channels going to the actuator it may also decrease the control performance of the system. Since the bore to the pilot stage is only connected to one side of the main valve, the DFCU can be designed in such a way that there is no leaking from the lines connected to the actuator. A simple example hydraulic circuit is displayed in figure 5.6. The bore from the pilot stage to the main stage is displayed as a small throttle in the circuit. In the example system the leaking causes only small power losses since some fluid leaks constantly from the main supply pressure to the pilot stage. However, the control performance of the system is not affected because both actuator lines are connected to the flow channel below the orifice of the main valve and therefore the channels are sealed tightly with the poppet.

The leakage could be prevented with an O-ring seal around the stem but it would complicate the structure significantly. Soft seals also wear in time which is not a desirable property. Some kind of membrane between the pilot and main stages is also a possibility but membranes are often problematic. Membranes cannot often endure large pressure differences over them, they cannot transmit large movements in a small space and they usually have a large surface area which translates to large pressure forces acting on them. Leaking between the main and the pilot stages can be reduced also with very tight tolerances but this brings problems to the manufacturing. Therefore if there is a pressure difference between the pilot stage and the main stage there will also be some leaking between them.

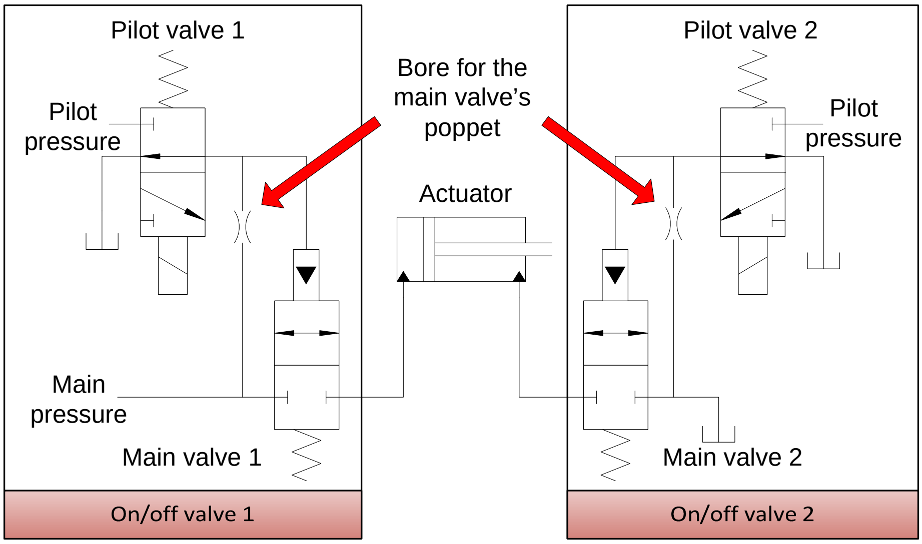


Figure 5.6: A simple hydraulic circuit showing the flow path of the leaking between the pilot and main stages as a throttle.

5.2.2 Return spring and piston

The pressure force acting on the main poppet was calculated to be about 29 N with 20 MPa pressure in chapter 3.1. However, if the higher pressure is on the upper side of the main valve's orifice, the pressure force is acting on the whole cross sectional area of the poppet's stem instead of only the area of the orifice. To ensure that the poppet seals the orifice properly, the diameter of the poppet's stem (D_s in figure 3.1) was decided to be 0,3 mm larger than the diameter of the orifice (D_o in figure 3.1). Therefore the maximum pressure force acting on the stem and thus on the piston is 42,8 N when calculated with equation (3.3) and a stem diameter of 1,65 mm. Actually the diameter of the stem was rounded later to 1,7 mm for manufacturing reasons which slightly increased the pressure force.

The simplest structure for the return spring and the piston of the main valve is a straight bore where they both move in, like displayed in figure 5.7. If the spring would be larger than the piston there would need to be a shoulder in the bore which would complicate the manufacturing. In chapter 3.2 was calculated that the optimal diameter for the piston of the main valve would be about 4 mm. However, a spring with such a small diameter that could produce enough force could not be found. The smallest suitable spring had a diameter of 6 mm and the maximum force

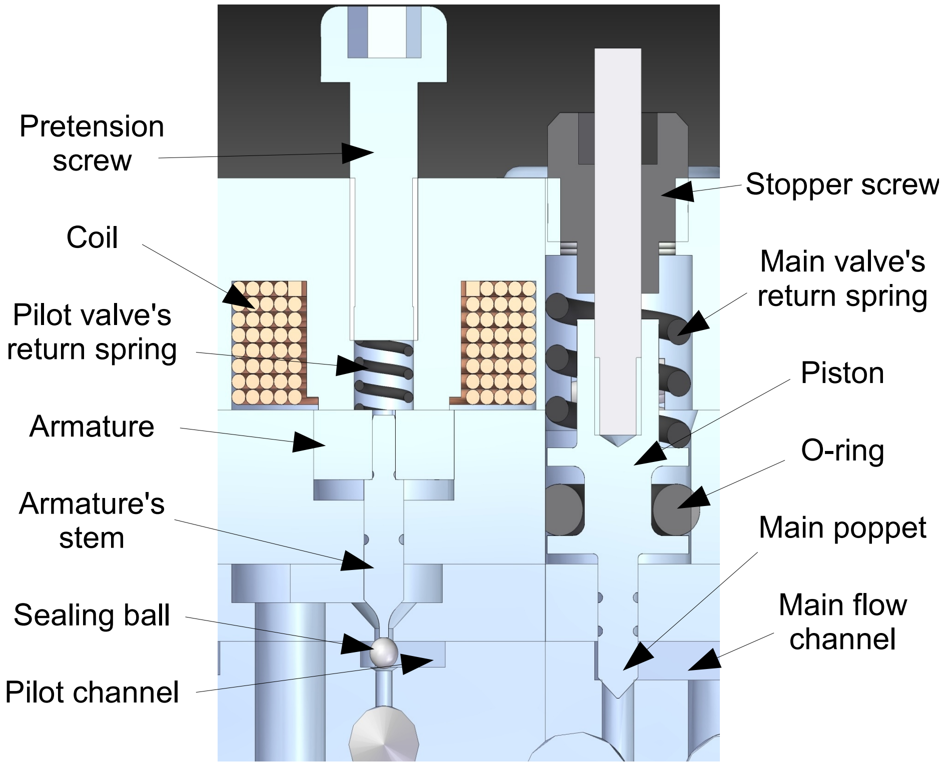


Figure 5.7: A cutout of the pilot valve and the piston of the main valve.

it could produce was 44,5 N. The diameter of the piston was selected to be the same as the diameter of the spring so both the piston and the spring can move in a simple straight bore. There is an O-ring in a groove around the piston to separate the different sides of the piston.

5.2.3 Layout

The outer diameter of the solenoid's coil is 13 mm but the diameter of the main valve's return spring is only 6 mm. Because both the return spring and the solenoid are quite tall compared to the total thickness of the valve structure but their diameter is very different, the most efficient way in terms of space usage is to place them side by side in the structure. In figure 5.8 is displayed three ways to arrange the four coils and main valve's return springs in a four-valve DFCU. If all of the coils in the DFCU would be positioned similarly with the main valve's spring next to each other like in the layout on the left side of the figure, lots of space would not be used efficiently. The layout displayed in the middle of the figure is more

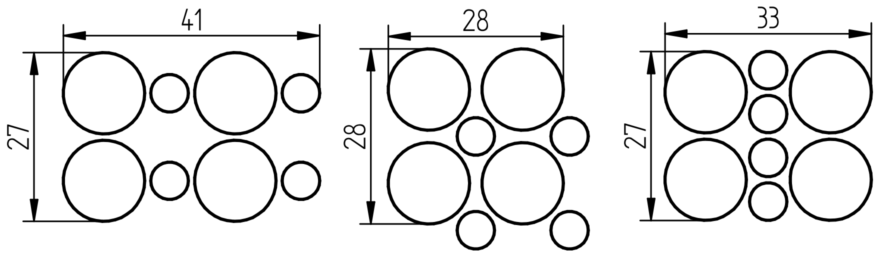


Figure 5.8: Different ways to arrange the coils and the main valve's return springs.

efficient. The layout on the right is not as efficient as the one in the middle, however in it all the main valve's poppets are in one row. This lets all of the valves use only one drilled flow channel which makes the machining easier. Therefore the layout on the right was chosen for the prototype.

The height of a single valve is approximately 20 mm in the designed prototype. From the figure 5.8 can be calculated that one valve requires approximately 2,23 cm^3 of area in the prototype. Therefore we can calculate that the volume of the designed valve is about 4,5 cm^3 without taking into account the main flow channels, control electronics etc. The outer dimensions of the body of the DFCU prototype are 70 x 70 x 38 mm.

5.2.4 Settings and measuring

In the top layer of the DFCU prototype there are similar screws as in the actuator prototype for adjusting the pretension of the return springs of the pilot valves. There are also screws for adjusting the openings of the main valves. These screws are visible in figure 5.7. The stopper screw for the main valve limits the upward movement of the piston by creating a mechanical stop with an adjustable position. The same screw also adjusts the pretension of the return spring of the main valve. The opening and the pretension can be adjusted individually only by changing the screw to a different one. By changing the dimensions of the shoulder machined to the screw, the compression of the return spring can be adjusted while keeping the opening of the valve same. When the opening of the valve is increased by moving the stopper screw, also the pretension of the return spring is reduced which slightly increases the response time with larger openings.

In the figure 5.7 there is also visible a rod which protrudes out from the top of the prototype through the stopper screw. This rod is attached to the main valve's piston. This way the position of the piston can be measured easily from outside of the prototype. Otherwise the response time of the valve would have to be measured from the changes in the pressure in the main flow channels.

The position or the movement of the armature, or the pilot valve's sealing ball, cannot be measured in the prototype. The structure is very small so there is not much space for a position sensor. Also the force generated by the solenoid is quite small and therefore a similar measurement system as was built for the main valve's piston might interfere too much with the solenoid's operation.

There are four small pressure transducers mounted to the bottom layer of the DFCU. Two of the transducers are for measuring the pressures from the main flow channels on the upper and the lower sides of the main valve's orifices. The other two transducers are for measuring the supply and the tank pressures for the pilot stage. The transducers are piezoresistive and their model is Keller 4 L.

5.2.5 Geometry of the poppets and the seats

The geometries of the poppet and the seat have a significant effect on the flow coefficient of the valve and also on the valve's sensitivity to cavitation. According to Heino [16], in water hydraulic seat valves a chamfered seat results to a better flow coefficient than a sharp cornered seat but a chamfered seat also cavitates more easily. All in all Heino concluded that a sharp cornered seat is preferable to a chamfered seat. Because a sharp cornered seat is also easier to machine it was chosen for the first prototype. The diameter of the orifice in the main valve is 1,35 mm.

According to Heino, the geometry of the poppet has no practical effect on the flow and cavitation characteristics of the valve. However the effect of the angle on a conical poppet's head was simulated by Karvonen et al. [9] and they concluded that at least a very sharp or a very blunt poppet is not practical. A sharp poppet requires a larger opening for the valve to achieve the same flow rate than a blunt poppet. A blunt poppet on the other hand seems to require more force to open the valve. Therefore a 90° angle ($\alpha = 45°$) was chosen to the poppets in the DFCU prototype.

The geometry of the pilot valve is defined by its structure and manufacturing aspects. The upper orifice has a sharp edged seat since the seat is a part of the third layer from the top in the DFCU, which is only machined from the upper side. The lower orifice must be chamfered so that the sealing ball is not pushed away by flow forces but instead it stays trapped between the two seats. The lower orifice of the pilot valve has a diameter of 0,7 mm and the upper orifice 0,8 mm.

5.2.6 Manufacturing

The main layers for the valve prototype were machined in a specialized company because of the tight tolerance requirements. There are several holes in the layers which have to be very concentric. For example the piston moves in a bore in the second layer from the top but the stem of the piston moves in a bore through the third layer from the top. Because the piston and the stem are features on a single part, these bores have to be very concentric or otherwise the part does not fit in them. Furthermore the main valve's orifice is in the bottom block and it also has to be concentric to the bores of the stem and the piston. This sounds very difficult to achieve but after all the structure worked very well. The layers were positioned very accurately with respect to each other with the help of four alignment pins. There are four concentric bores through all the layers of the prototype where the pins fit firmly. This forces also all the other bores in the layers precisely to right position. The main parts of the DFCU prototype are displayed assembled in figure 5.9. The bores for the centering pins are visible in each corner of the prototype.

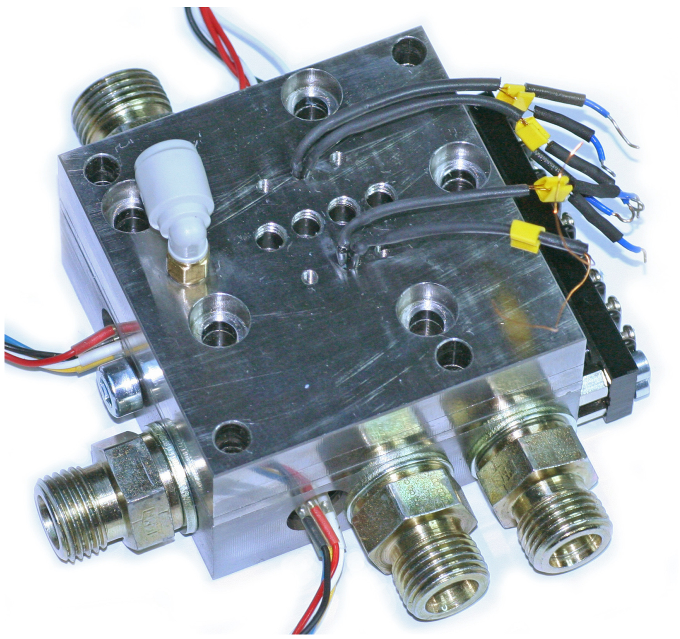


Figure 5.9: The main parts of the DFCU prototype assembled.

The stems of the armature and the pistons were turned in our own laboratory. Neither of them was hardened. A piston with an O-ring, return spring and a stopper screw are displayed in the left in figure 5.10. On the right is displayed an armature with a stem attached, its return spring and a sealing ball. The stems of the armature have a flat surface in touch with the sealing ball. The flat surface was assumed to be large enough compared to the forces involved so it would not deform significantly. The surface pressure from the seat on the main valve poppet was approximated and the poppet was assumed to deform slightly. Both the poppet's and the armature's stems have a groove around them for centering and lubricating the stem better when it moves inside the bore. The groove lets the fluid distribute evenly in all sides of the stems.

(a) Main valve's poppet with the piston, stem for measuring movement, return spring and the stopper screw.

(b) The armature, stem, return spring and sealing ball.

Figure 5.10: Small parts of the valve. The diameter of the armature and the piston is 6 mm.

Because the armature in the actuator prototype was fairly difficult to manufacture, a different method was designed for the DFCU prototype. It was decided to machine the armatures with a wire electric discharge machine (Wire EDM). A CAD model of the armature designed for Wire EDM is displayed in figure 5.11. Wire EDM can cut the armature or multiple armatures from a metal sheet easily and precisely. Also small and complex features are easy to cut with wire EDM. All the features in the armature are made with one cut so there needs to be a narrow cut to the attachment hole in the center. The width of the cut is less than 0,4 mm so it does not have a significant effect on the surface area of the armature. The flow channels for the fluid displaced by the armature's movement are cut to the sides of the armature.

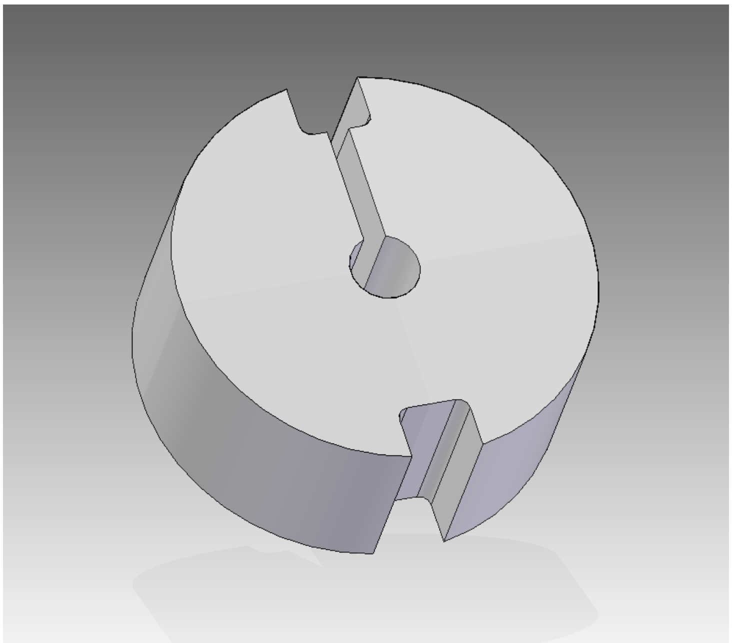


Figure 5.11: The CAD model of the armature designed for wire EDM.

Chapter 6

Testing and results

6.1 Actuator prototype

To keep the actuator prototype stationary while testing, it was mounted horizontally on a steel plate. The mounted actuator is displayed in figure 6.1. In the figure is visible the top side of the prototype. The armature's movement was measured with NAIS LM300 laser distance sensor from the stem that protrudes out of the prototypes bottom. The response time of the sensor is 0,1 ms which was just enough for the tests. The analog output signal from the laser sensor as well as the current in the coil were recorded with Fluke Scopemeter portable oscilloscope.

Normally the solenoid is a part of the valve structure which is filled with fluid. During the tests the bore, where the armature moves in, was manually filled with hydraulic oil to simulate real conditions. If the bore would be only partly filled with oil the armature would move significantly faster as the viscous friction would not slow it down so much. Oil was injected to the bore via the red connector visible in figure 6.1.

Input voltage to the coil was controlled by Apex MP108FD power operational amplifier. A separate microcontroller board was designed and built for controlling the power operational amplifier. The microcontroller produces a control signal with its digital to analog converter which is then processed by an analog circuit on the same board and used to control the power operational amplifier. The analog circuit is basically a P-controller with a feedback from the current in the coil. This setup made it easy to test any desired current levels because the desired current curve could be defined with the microcontroller. The input voltage to the operational amplifier was $\pm$ 24 V and the output voltage to the coil was a couple of volts smaller. Output current of the amplifier was limited to 7 amperes.

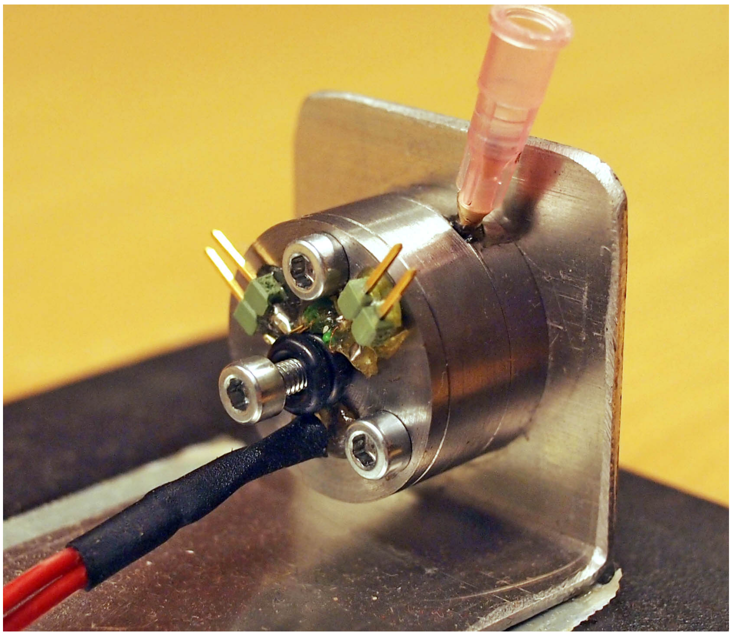


Figure 6.1: Actuator prototype mounted for testing.

The solenoid was also tested with a self-made PWM based controller with some larger input voltages. Input voltages of over 24 volts however did not give much advantage over lower voltages in the response times. This is probably due to the quite low number of loops in the coil which makes its inductance low. Therefore the current rises already quite fast in the coil and the response time of the actuator cannot be improved significantly by improving the response time of the current.

Figure 6.2 presents the measured actuator′s coil current and the position of the armature. The pretension of the return spring was adjusted to get similar opening and closing response times. The actuator seemed to produce the best results when the spring was compressed almost fully when the armature was pulled in by the solenoid. This way the spring produces the most return force. The spring force was measured to be roughly 9 N. Therefore it can be concluded that the solenoid also produces significantly over 10 N of force since it can overcome the spring force and also pull the armature in very fast. The required voltage to keep the solenoid closed was 0,38 V and the current was measured to be 0,6 A. Thus the required holding power was about 0,23 W.

As can be seen from the figure 6.2 the armature starts to move about 0,5 ms after the current starts to rise in the coil. This is very close to the

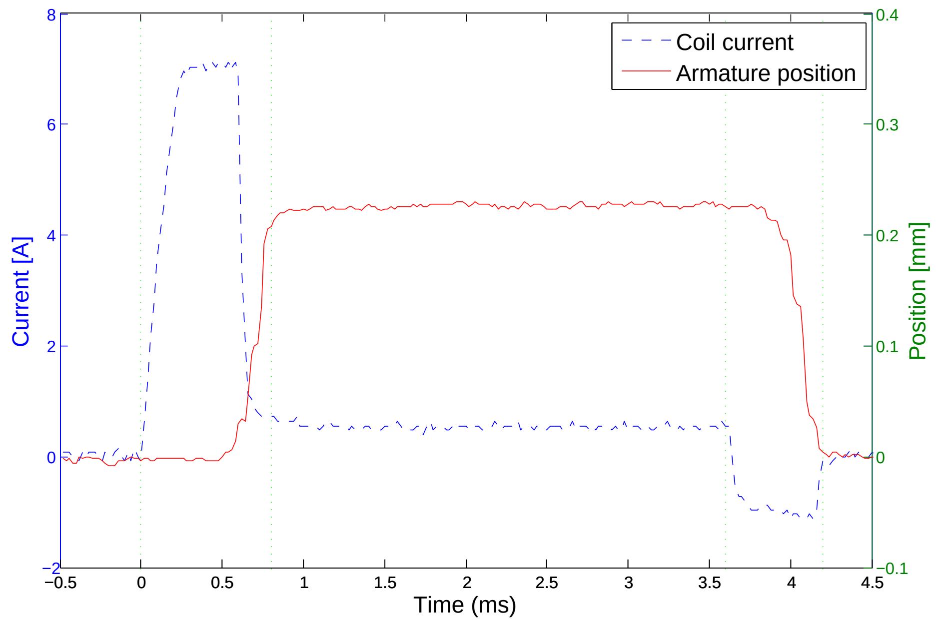


Figure 6.2: Actuator prototype's measured coil current and position of the armature.

response simulated in chapter 4.2. The armature has moved the desired 0,2 mm at about t = 0,8 ms and therefore the opening response time of the actuator is about 0,8 ms. The movement of the armature takes considerably less time than what was simulated with the Simulink-model presented in chapter 4.2. This is probably because the solenoid produces more force than what the FEM simulations with quite conservative current inputs predicted. The closing response time was measured to be 0,6 ms.

The factors contributing to the delays in the operation of a solenoid actuator are listed by Brauer to be: [2, p.125].

1. Delays in the control electronics supplying voltage and current
2. Rising of the coil current delayed by inductance
3. Rising of the magnetic flux delayed by eddy currents
4. The force generated by the solenoid rises as the magnetic flux rises
5. The force produces acceleration to the armature and any attached mass
6. After a certain time the armature reaches the end of its stroke

The delays caused by the switches in the control electronics are negligible compared to the other delays in the actuator. From figure 6.2 can be seen that in this prototype the current in the coil rises to its maximum value in about 0,2 ms. The armature starts to move about 0,5 ms after the current starts to rise so it takes 0,5 ms for the solenoid to generate enough force to overcome the force of the return spring. Part of this delay is caused by the delayed rise of the current but mostly it is caused by the eddy currents in the magnetic circuit. The movement of the armature takes about 0,3 ms which is less than half of the total response time of the solenoid. Therefore it can be concluded that eddy currents cause the major part of the response time of the actuator.

6.2 First tests of the valve prototype

The DFCU prototype was tested on two occasions. The first time only one valve at a time was tested since no control electronics for all of the valves was available. A picture of the DFCU in the test setup is displayed in figure 6.3. The same electronics was used for controlling the solenoids as when testing the actuator prototype. National Instruments USB-6125 data acquisition (DAQ) device was used with a laptop for measuring. The laptop was running a measuring program created with Labview, which reads and stores values from the DAQ device and relays commands from the user interface via serial bus to the microcontroller controlling the operational amplifier.

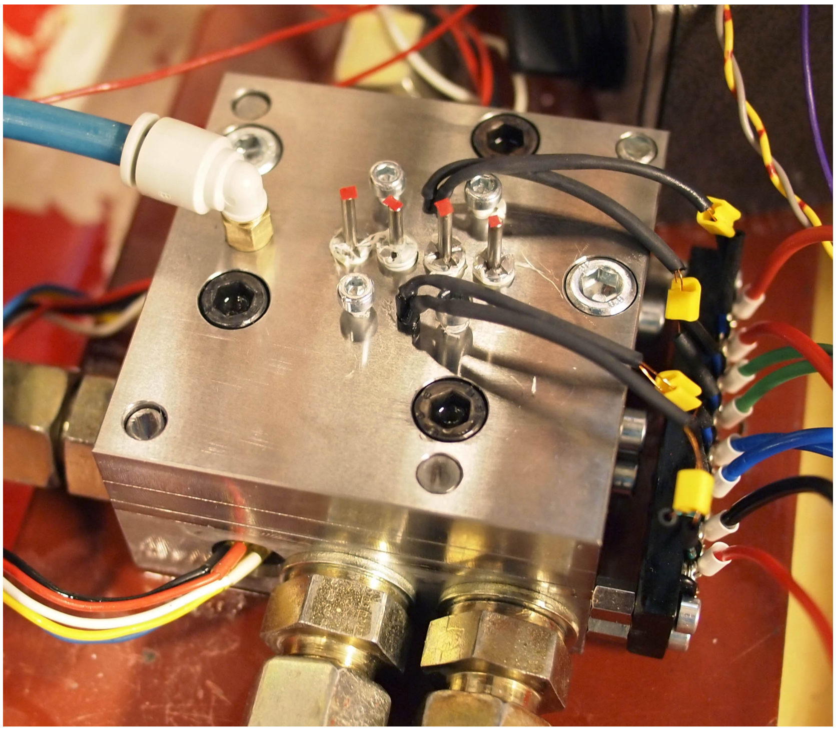


Figure 6.3: DFCU prototype in the test setup.

The hydraulic circuit of the test setup is displayed in figure 6.4 and its parts are annotated below the figure. The hydraulic part of the test setup for the valve prototype consisted of a main hydraulic power unit, a separate small power unit for pilot pressure, a 3/2 directional valve and a flow meter. The hydraulic fluid used in the tests was Mobil DTE Excel 68 hydraulic oil. The oil was approximately room temperature in the tests since the power unit was running only intermittently and the power it a very large tank. The flow meter was Kracht VC0.4 which is a gear type meter for flows from 0,2 l/min to 40 l/min. Flow rate to or from the pilot stage was not measured since it was expected to be very small and no flow meters for measuring such small flows were readily available in our laboratory.

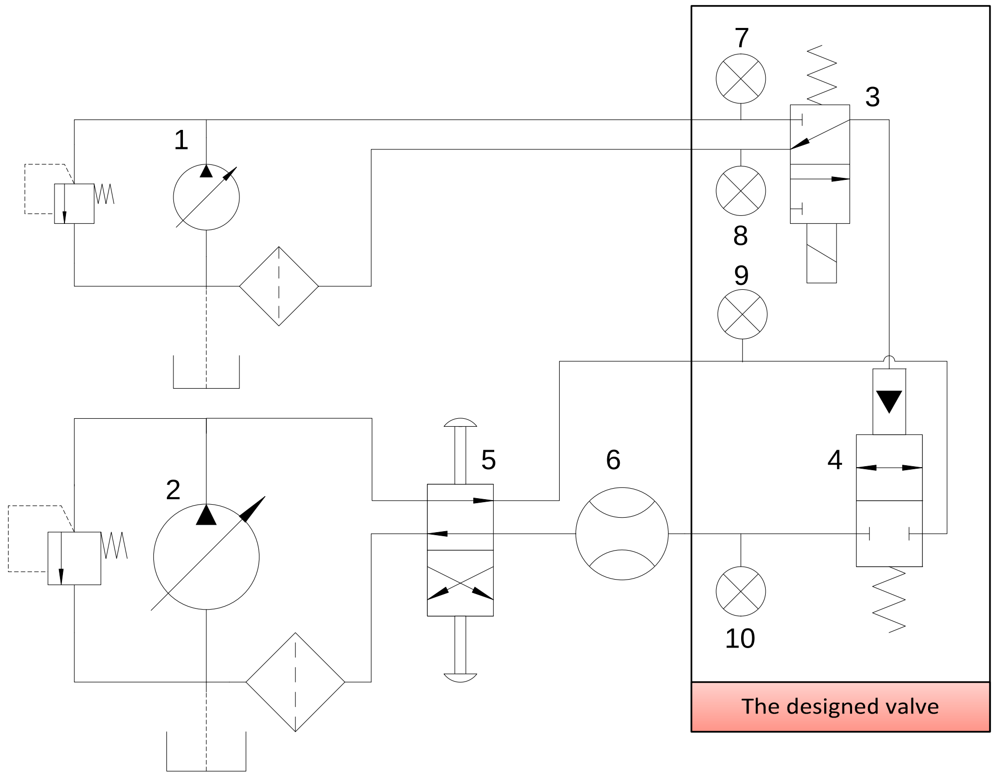


Figure 6.4: The hydraulic circuit of the test setup.

1. Power unit for the pilot pressure
2. Power unit for the main pressure
3. Pilot valve
4. Main valve
5. Directional valve
6. Flow meter
7. & 8. Pressure transducer for the pilot pressure
9. & 10. Pressure transducer for the main pressure

6.2.1 Flow characteristics

The flow characteristics of the valves were measured by keeping one of the valves open and then adjusting the pressure difference over the valve slowly from about 0,4 MPa to 25 MPa. Then the direction of the pressure difference was reversed with the directional valve and a similar pressure sweep was made. Flow was measured with valve openings of 0,4, 0,6 and 0,8 mm. The pressure difference over the valves was measured from inside the DFCU prototype with the two pressure transducers placed on different sides of the valves.

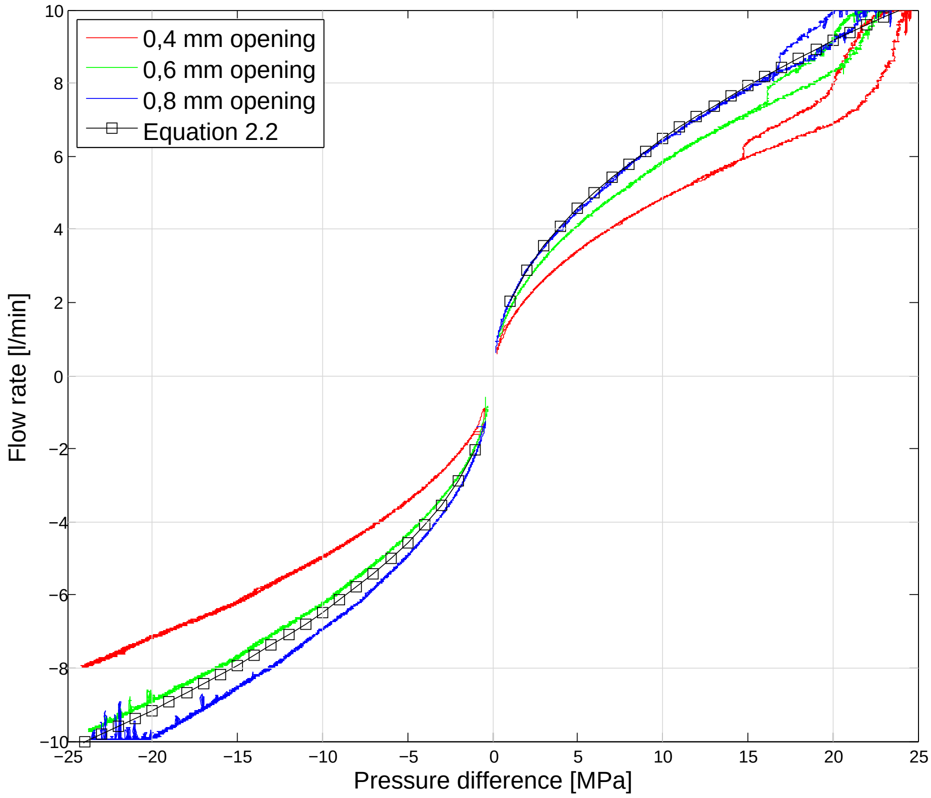


Figure 6.5: The characteristic curves of valve number 3 of the DFCU prototype with three different openings.

The characteristic curves in figure 6.5 were made from the measured flow in liters per minute with respect to the pressure difference in MPa. The pressure difference is defined negative when the higher pressure is on the upper side of the orifice i.e. the flow in the main valve is converging. From the figure can be seen that the shape of the measured flow rate curves follow very closely the equation (3.2) for a conical orifice with a 0,4 mm opening and a flow constant of 0,6. However, the measured flow rate is significantly lower.

From figure 6.5 can also be seen that the flow rate starts to increase rapidly after over 20 MPa positive pressure difference over the valve. This is because the pressure force on the poppet exceeds the spring force of the main valve's return spring. Therefore also the three other valves, in addition to the valve currently tested, open and start passing flow through them. There is also hysteresis visible when the other valves are opened by the pressure. This is because when the valves are closed, the pressure acts against the surface of the poppet which is closing the orifice, but when the valves open, the pressure acts on the whole cross sectional area of the poppet's stem and thus a smaller pressure is enough to keep the valves open than what is required to open them. When the pressure difference is negative, the pressure pushes the poppet harder against the seat and thus this problem does not occur.

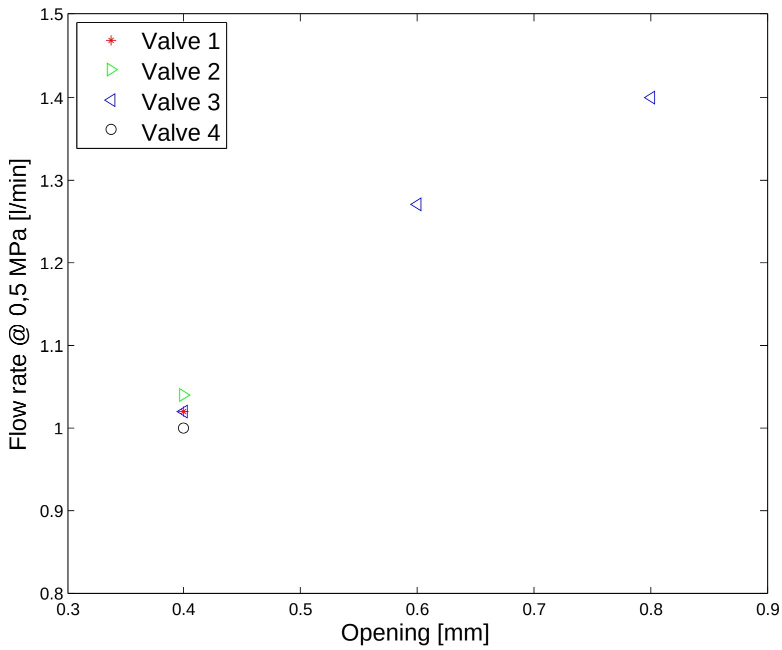


Figure 6.6: Flow rates measured with 0,5 MPa pressure difference.

The flow rate requirement of 1,4 l/min for the designed valve was set for a 0,5 MPa pressure difference. Figure 6.6 shows the measured flow rates with a 0,5 MPa pressure difference with respect to the opening of the valve. All of the valves in the DFCU prototype were not tested with all the openings. The valve number 3 has the most measured data from the first tests and it is expected to represent also the other valves well. From the figure 6.6 can be seen that with the initially designed opening of 0,4 mm the flow rate is only about 1,0 l/min. Only when the opening is increased to 0,8 mm the flow rate reaches the designed 1,4 l/min. This is considerably less than what was calculated in section 5.2 with two different equations. The analytical equations for flow rates are at best quite vague since the flow coefficient has a significant effect on the results.

6.2.2 Response time

Response times were also measured with 0,4, 0,6 and 0,8 mm openings. The response time was measured by setting the valve to switch on and off every 0,7 seconds. At the same time the whole pressure range was swept over the same way as when testing the flow rate. Pressure difference over the valve, current in the coil and the location of the poppet of the main valve were logged during the switching events and the log was analyzed afterwards with a Matlab script. The beginning of a switching event was determined as a noticeable change in the current. This occurs only a few microseconds after the microcontroller switches the boost voltage on. The end of a switching event was considered to be the moment when the poppet has traveled 90 % of its travel distance i.e. the valve is 90 % open. By calculating the duration of several switching events and plotting them as a function of the pressure difference we get figure 6.7.

From figure 6.7 can be seen that the opening response time of the valve is almost independent from the pressure difference over the valve, varying only some tenths of a millisecond. The opening distance of the valve obviously affects the opening response time slightly because the poppet has to move further, but the opening response times stay between 1,5 and 2 ms. On the other hand the closing response time is strongly dependent on the pressure difference. This is because a large pressure at the main valve's orifice creates a large pressure force which pushes the poppet upwards. This pressure force acts against the return spring slowing down the return movement of the poppet. If the pressure difference is large enough the pressure force prevents the movement completely. The slowing effect doesn't seem to depend much on the direction of the pressure difference across the main valve.

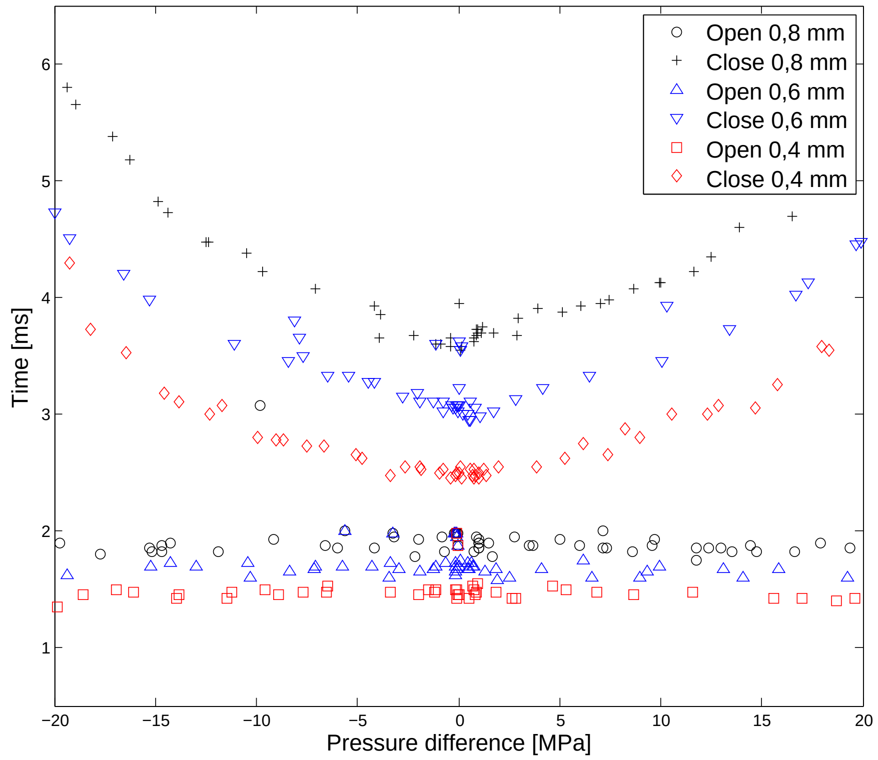


Figure 6.7: Response times of the valve 3 with different openings and pressure differences over the valve in the first tests.

6.2.3 Discussion

From the fact that the valves do not stay closed with a large positive pressure difference and that the closing response time is too slow and strongly pressure dependent can be concluded that the return springs of the main valves are too weak. The chosen coil springs were the stiffest springs of their size that could be found at the design phase, but based on the tests they were not stiff enough. The maximum force at the maximum deflection of the current springs is 44,5 newtons. With further searching could be found coil springs with a similar outer diameter but slightly thicker wire. The springs were too long for the prototype but they could be cut shorter. The maximum force they could produce was 72 newtons which is over 60 % more than the former springs. Coil springs should however not be stressed to their maximum strain when they are expected to endure millions of operating cycles. Even though the new springs would provide enough force to reduce response time and increase the pressure range they would be stressed to their maximum and probably fail too fast.

This is why also disc springs were considered as a replacement for the current springs. Disc springs are very stiff compared to their size but their maximum deflection is small. However, several disc springs can be stacked together to get more deflection while still generating the same force. Because the bore for the return spring is designed for a coil spring it is very high compared to its diameter. The smallest stock size for disc springs seems to be usually 6 mm in diameter which does fit into the bore. Maximum force for this size disc springs is over 100 N. The usable deflection with a proper preload for this size disc spring is only about 0,05 mm. This means that a 16-disc-stack is needed to get enough deflection and force at the same time. The whole stack still fits into the same space as the previously used coil spring.

A second problem during the testing of the DFCU prototype was significant external leaking. The leaks originated mainly from the five drilled flow channels which were sealed with set screws. The set screws apparently did not seal the bores with the cones in their tip but instead the hydraulic fluid leaked constantly through their threads. Also some leaking originated from between the layers and through the threads of the adjustment screws for the pilot valve's return springs. After the first tests the set screws were removed and the flow channels were sealed with socket head screws which were modified in a lathe to have a cutting edge under the shoulder in their head. The edge cuts into the block sealing not from the threads but from the edge.

After the tests the stem of the pilot valve and the poppet of the main valve were inspected for deformations. The main valve's poppet had a small groove around it. The depth of the groove was only a few hundredths of a millimeter. Surprisingly the stem of the pilot valve had a much more significant deformation than the main valve's poppet. Tip of the stem after the first tests is displayed in figure 6.8. Originally the tip of the stem was rectangular but in the figure it is clear that the tip has widened due to the stress. The deformation may be a result of the normal operation of the valve or it may also be caused by accidental over tightening of the return spring's pretension screw. Whatever the reason for the deformation was, the stem as well as the poppet should be hardened in the following prototypes. The stem can be hardened without any problems since it is not a part of the magnetic circuit.

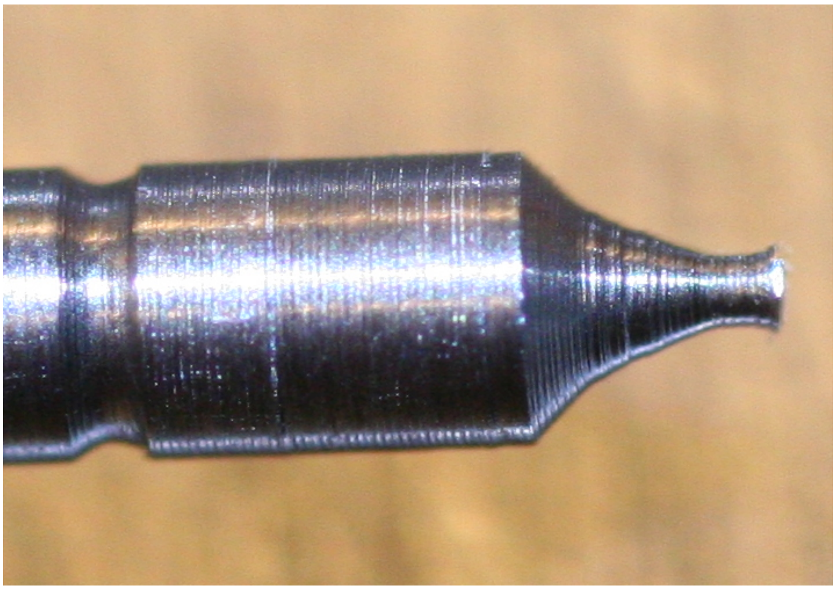


Figure 6.8: The tip of the pilot valve's stem after the first tests.

6.3 Second tests of the valve prototype

The test setup for the second tests was similar to the first tests except for the control electronics of the valves and the return springs of the main valves. A new electronics board, able to control all the four valves simultaneously, was designed and built for the tests. The board is provided with two voltage levels from a laboratory power supply, one for the boost and one for the holding current, which are then switched to the coils when required. With the new electronics it was not possible to measure coil currents with the DAQ device used in the tests, instead the control signal from the microcontroller to the electronics was considered the beginning of a switching event when determining the response times.

Because the return springs of the main valves were determined to be too weak, they were changed to stiffer springs for the second tests. Two of the valves were fitted with stiffer coil springs and the two others were fitted with 16-disc-stacks of disc springs. The coil springs produced a maximum force of 72 N and the disc springs produced a force of 116 N.

The flow rates of the valves did not change compared to the first tests, however the unwanted opening of the valves with high positive pressure difference disappeared. None of the valves opened on their own even with a 25 MPa pressure difference over them. No higher pressure was

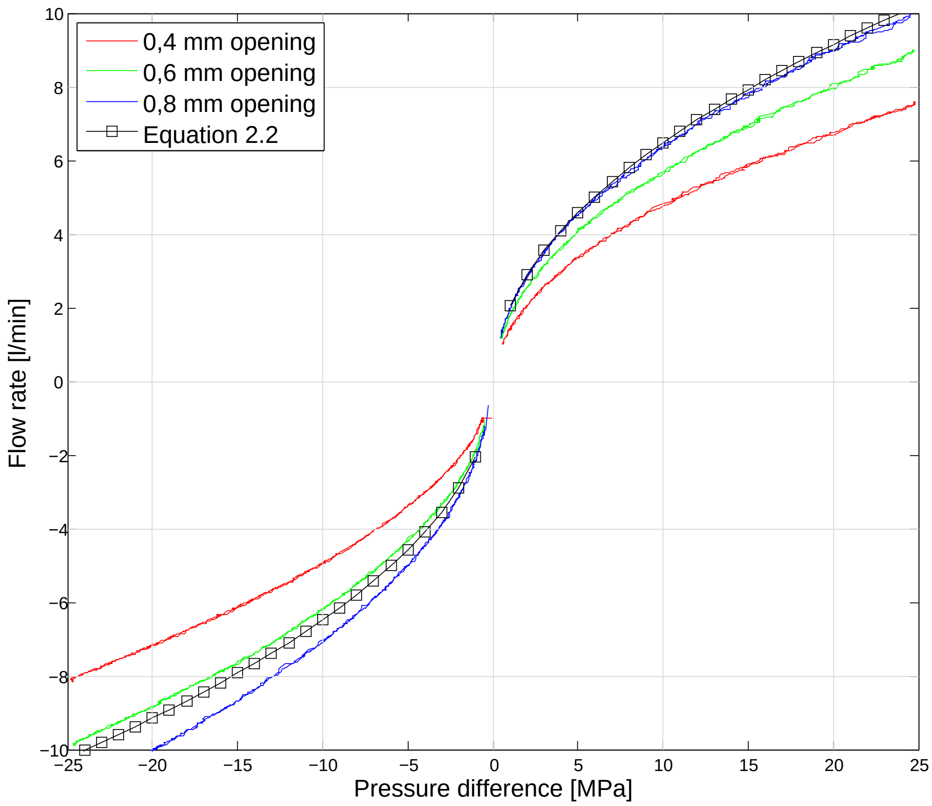


Figure 6.9: The characteristic curve of the valve 4 in the second tests.

tested since the pressure transducers installed in the package were only meant for a pressure range from 0 to 20 MPa. They however seemed to be measuring accurately up to 25 MPa. Characteristic curve of valve number 4 measured in the second tests is displayed in figure 6.9.

The closing response times of the valves improved significantly over the previous tests. The valves with the stiffer coil springs had closing response times of 2,5 to 4 ms, instead of the previous 2,5 to 6 ms, depending on the pressure difference. However, the disc springs improved the response times even further. The response times of valve 4 with disc springs are displayed in figure 6.10. With the disc springs the opening and closing response times with a 0,4 mm opening and a small pressure difference over the valve became symmetric i.e. they both were 1,2 to 1,3 ms. With a larger pressure difference the closing response time rose up to 1,7 ms.

With larger openings the opening response time increased only a little,

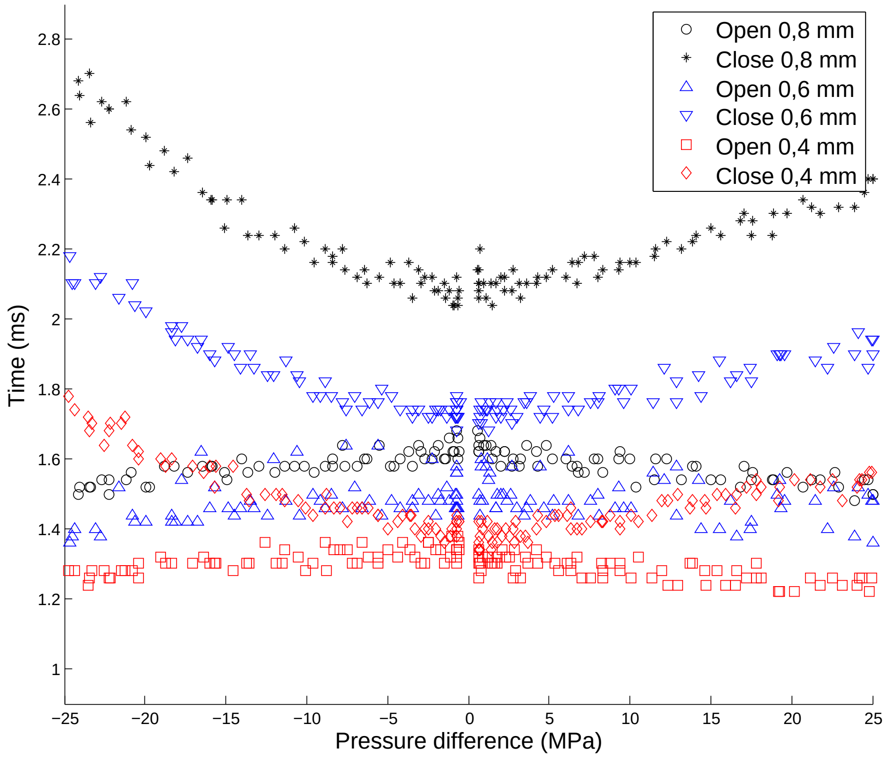


Figure 6.10: Response times of the valve 4 with different openings and pressure differences over the valve in the second tests.

up to 1,6 ms at with 0,8 mm opening. The closing response time was again much more affected by the opening increasing up to 2,1 ms with 0,6 mm opening and up to 2,7 ms with 0,8 mm opening. With a 0,6 mm opening also the closing response time stays under the desired 2 ms with the -20 to 20 MPa operating pressure required from the DFCU.

It was not possible to directly measure the movement of the armature or the pilot valve's stem. The response time of the pilot valve or the solenoid can however be approximated by studying the delay between the opening command and the moment when the main valve's piston starts to move. At that moment the pilot valve is at least partly opened. The position of the piston with respect to time during a 0,6 mm opening cycle of one valve is displayed in figure 6.11. From the figure can be seen that the piston starts to move about 0,7 milliseconds after voltage is switched to

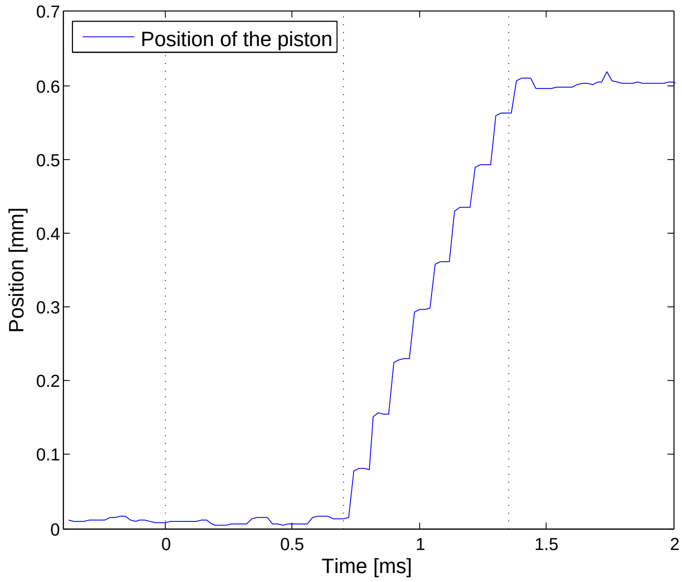


Figure 6.11: The opening response of the main valve's piston with a 0,6 mm opening. The voltage to the coil is switched on at t = 0.

the coil. Therefore the response time of the pilot valve can be determined to be about 0,7 milliseconds. The movement of the piston is complete at about 1,35 ms and thus in this case the delay caused by the movement of the piston is almost the same as the delay caused by the pilot valve. In the figure 6.11 it is also visible that the limited sampling frequency of the laser distance sensor causes stepwise increments to the position measurement.

The response time of the main valve is dependent on the pressure in the main flow channel as well as the opening of the main valve. The response time of the pilot valve however does not depend on the properties of the main valve. Figures 6.12 and 6.13 were made by analyzing several opening and closing cycles of the valve in a similar way as described previously. Figure 6.12 displays the response times of the pilot valve and the main valve with respect to the pressure difference over the main valve during the opening cycle. In figure 6.12 the 50 kHz sampling frequency of the data acquisition hardware and the 12,5 kHz sampling frequency of the laser sensor cause the results to be aligned in rows. It can be seen that the response time of the pilot valve, displayed in red in the figure, is from 0,7 to 0,8 ms regardless of the pressure difference or the opening of the main valve. However the time it takes for the main valve's piston to move

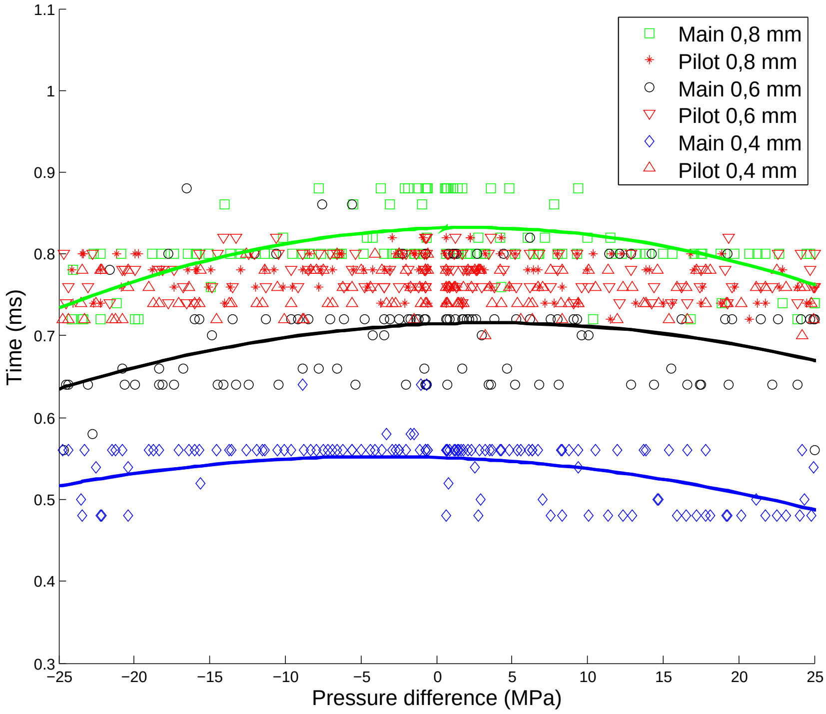


Figure 6.12: Opening response times of the pilot stage and the piston of valve 4 with different openings with respect to the pressure difference. Polynomial curves are fitted to the response times of the poppet.

depends strongly on the opening and also slightly on the pressure.

Figure 6.13 shows the response times of the pilot and the main stages during the closing cycle of the valve. Response time of the pilot valve is again 0,7 to 0,8 ms but the main stage's response time has large variations depending on the opening and the pressure difference. From figures 6.12 and 6.13 can be concluded that the delay caused by the solenoid causes the main part of the delay in the whole valve's response when the opening of the main valve is about 0,5 mm or less. With larger openings the travel time of the piston is a more significant factor especially during the closing cycle.

Interestingly also the opening response time of the valve seemed to improve slightly compared to the first tests even though the stiffer springs should have slowed down the opening movement of the piston. By analyzing the data from the first test it was concluded that the response time of the solenoid actuator was in the first tests about 1 to 1,2 ms instead of the

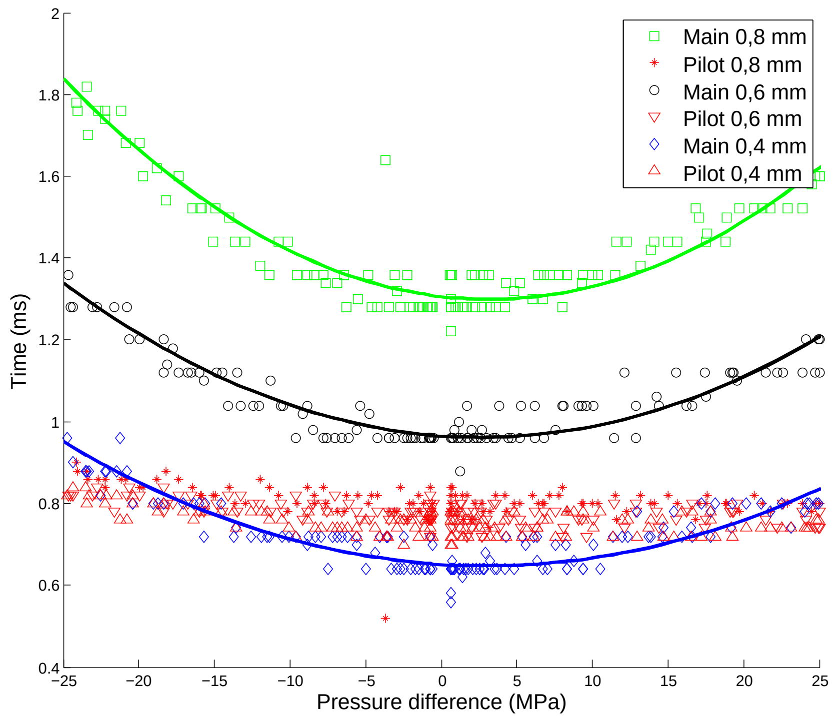


Figure 6.13: Closing response times of the pilot stage and piston of valve 4 with different openings with respect to the pressure difference. Polynomial curves are fitted to the response times of the poppet.

0,7 to 0,8 ms in the second tests. There are at least two factors which may have contributed to the difference in the response time of the solenoid. First is that even though the input voltage for the control electronics of the solenoids were the same in the two tests, the maximum output voltage to the coil was higher in the second tests. As mentioned in chapter 6.1 the operational amplifier in the first tests outputs a couple of volts less than the input voltage. The electronics in the second test have a smaller voltage drop over them which probably slightly improves the response time of the solenoid. Also with the electronics in the second tests the current was not limited to 7 A and it rose to about 9 amperes with the coils used in the prototype.

There are also several settings in the prototype that can be adjusted while testing. The opening of the main valve and the pretensions of the return springs for the pilot and main valves can be adjusted with screws. Also the length of the positive and negative boost pulses can be adjusted with both electronics. These settings were not recorded in the first tests and they probably were more optimized in the second tests which slightly improved the response times.

Chapter 7

Discussion

7.1 Comparison to other digital valves

As explained in chapter 2, valves specially designed for DFCUs are not currently commercially available. Therefore the designed valve is here compared mainly to two different valves developed at IHA, the hammer valve [30] and the Proto⌀10 [9][10]. The most important properties of the valves are displayed in table 7.1. The properties for the designed valve are displayed for a 0,6 mm opening.

In the table are also displayed the properties for the smallest commercial seat type hydraulic valve found, 250 Series solenoid valve from The Lee Co [28]. The response time specifications given by The Lee Co are not very specific but most probably they are measured without any boosting and therefore the response time could be improved considerably. Also the properties for Hydac WS08W-01, a direct operated solenoid actuated seat valve, are displayed. The Hydac WS08W-1 is a commercial cartridge valve which has been used in building DFCUs in some previous projects at Aalto [8]. The response time of the Hydac valve was measured in Helsinki University of Technology with a 24 V boost voltage. The measurements were made with dry valves i.e. the viscous friction of the fluid or any pressure forces did not slow down the response.

It can be seen from the table that the designed valve is not quite as fast as the Proto⌀10 which is direct operated. However the response time is slightly faster than that of the hammer valve and compared to these prototypes the commercial valves are much slower. The flow rate of the designed valve is about six fold compared to the Proto⌀10 valve or the Lee 250. On the other hand the flow rate of the hammer valve with its clever but difficult to manufacture poppet, is almost double compared to

	Designed valve	Proto⌀10	Hammer	Lee 250	Hydac
Response time [ms]	1,4 - 2,0	1,2 - 1,5	~2	<15 - 40	~12
Maximum pressure [MPa]	>25	20	21	21	25
Flow rate @ 1 MPa [l/min]	1,8	0,3	3,3	$\sim 0,3$	?
Max. flow rate q_{max} [l/min]	9	1,2	17	1,17	19
Bidirectional	Yes	No	No	No	No
Volume V [cm^3]	4,5	2,4	7	4,4	73
q_{max} / V [l/min/cm^3]	2	0,5	2,4	0,27	0,26

Table 7.1: The designed valve compared to two other prototypes and two commercial solenoid valves.

the designed valve. The size of the designed valve is about 65 % of the size of the hammer valve and almost twice the size of the Proto⌀10. An important property of the valve is the ratio between its maximum flow and its volume which can be considered flow density. The ratio for the designed valve is very close to the hammer valve and several times larger than for the Proto⌀10 or the commercial valves.

Therefore it can it can be concluded that with the designed valve can be built a DFCU with almost the same flow density as with the hammer valve but with a higher resolution and possibly easier manufacturing. The flow density with the current valve would be fourfold compared to a DFCU built only with Proto⌀10 valves, but the Proto⌀10 valves would provide slightly a better response time and better resolution. Compared to the commercial valves the designed valve is clearly superior in any measure. Therefore it is probable that in the future the best results will provide a DFCU which combines direct operated valves for even faster response time and improved resolution with pilot operated valves for improved flow rate.

7.2 Things to improve

There are two important aspects where the design of this DFCU prototype could be improved. They are the generation of the return force for the poppet of the main valve and the internal leaking of the prototype. Improving the structure of the piston and the spring responsible for the actuation of the main valve requires some changes to the structure of the valve but internal leaking could possibly be fixed also in the current prototype.

The selection of the size of the return spring and the piston for the main valve were affected by several factors. The most important of these were the forces acting on the poppet, required response time, layout of the valve and the structure of the piston system. The layout and compactness of the DFCU prototype and especially the positioning of the coils and the pistons of the main valve were given too much value when determining the sizes of the return spring and the piston. The minimum size of the spring was determined by the required force and the available springs. The maximum size of the spring was limited by the desired compact layout for the DFCU prototype. The piston was also selected to be as large as the spring in order to have only a simple straight bore for them, even though it was not the optimal size according to the calculations. This caused the imbalance between the force of the return spring and the force generated by the pilot pressure acting on the piston, which in turn caused imbalance between the opening and closing response times of the valve.

Even though the return springs were changed to twice as stiff ones, the force generated by the piston was still very large compared to the force of the spring. A large piston can easily create enough force to overcome the spring force and any pressure forces. However the speed of the opening movement as well as the closing movement of the piston is limited by the flow rate through the pilot valve since the piston has to displace fluid as it moves. With a larger piston the amount of displaced fluid is larger and therefore the piston moves slower. This affects especially the closing movement since not only the amount of displaced oil is larger but also the pressure generated to the pilot channel by the return spring is smaller. Therefore also the flow rate through the pilot valve to tank line is smaller and the piston moves even slower. Thus in the following prototypes the piston should be designed so that it is optimally sized regardless of the size of the spring.

Also the issue with leaking should be addressed better in the following prototypes. In the current prototype there was some external leakage which was mostly fixed, however the internal leaking is more of a problem. Digital valves are built with poppet valves because the contact between the seat and the poppet is virtually leak-free so it is not acceptable that the structure allows leaking past the valve somewhere else. In the tests related to this thesis leaking was not considered much since the structure of this prototype is such that it should be assembled either by gluing or soldering to make it leak free. It was hoped that with a good enough surface roughness the leaking between the layers in the prototype would be minimal, but it turned out to be significant. Also with the current structure it would have been difficult to determine where the leaking was originating. The following prototypes should be designed from the start so that leak prevention and testing is taken into account.

Other slightly less important factors to consider are the material selection for the magnetic circuit and the manufacturability of the valve. The AISI 12L14 steel selected for the current prototype is a good choice for magnetic applications but there is still room for improvement. Especially the electrical conductivity of the material could be lower, which would lead to less eddy currents and thus a faster response time. It is left for the future research to determine how much effect the conductance has on the response time. There are also other factors to consider in the material selection such as corrosion resistance, machinability etc. Corrosion resistance is important especially if the valve is required to be used with water hydraulics.

Manufacturability is not that much of an issue in prototyping since most parts can be manufactured in small quantities with reasonable effort. However in the long run also these valves are designed to be used by the industry and there it is important that the device can be manufactured in large quantities cheaply and fast. This aspect was considered already in the beginning of the design phase as many features were designed to be made into a small number of parts instead of making a large number of small parts. On the other hand it may actually be that mass production of smaller and simpler parts would be the most economical way. The current prototype has also some quite tight tolerance requirements which may or may not be a problem in manufacturing.

7.3 Safety aspects

An important aspect in valve design is safety. The valves should always act in a predetermined manner also in fault situations. Usually this means that if the valve for some reason loses its electrical or hydraulic power supply or the control signal it should automatically close to prevent uncontrolled movements in the system. If the valve does not close without external power, as in the case of bistable valves, this feature can be accomplished with the help of some emergency power storage. Usually the valves are designed to be normally closed i.e. they have for example a return spring which closes the valve when power is lost. This is true for the valve package designed in this thesis because if the pilot stage loses electricity, the coil will not hold the solenoid open, spring will close the pilot valve and after that the main valve will be closed by a spring. If the valve package loses pressure supply the valves will still close automatically no matter what the control signals are since the pilot stage needs pressure to keep the main valve open.

Often it is also desirable that the system is still operational in a fault situation. Valves with a direct operated solenoid as their actuator are still operable when there is no supply pressure in the system. This is often useful for driving the system to a safe state in fault situation. For example a load can usually be lowered without external power to the system. This is not the case with pilot operated valves such as the valve designed in this thesis, since they need both electricity and a pressure supply to operate. For the applications where it is required to be able to operate the machine also when the pressure is lost, there could be some fail safe mechanisms built into the valve package. For example when a limited number of valve operation cycles are required in fault situation a pressure battery could provide the very small amount of flow the valve requires for operation. Another solution could be to equip the DFCU with a small electric motor running a miniature hydraulic pump. Best option would probably be to build the DFCU by combining direct operated and pilot operated valves.

7.4 Pilot operation vs direct operation

It is clear that pilot operated valves have both advantages and disadvantages. The most important advantage is that when compared to direct operated solenoid valves, or virtually any direct operated valves, pilot operation enables controlling a larger power with a same sized actuator. In practice this means that a larger flow rate, larger pressure or both can be controlled with a similar sized valve. Therefore a DFCU or a digital valve system with a large flow rate can be miniaturized to a small space. This is most important in mobile applications but also brings benefits in industrial applications.

On the other hand it may not be practical to miniaturize pilot operated valves even further to control very small orifices since then the pilot valve would become very small. This could cause problems with manufacturing tolerances and contaminants in the hydraulic fluid. Therefore if a DFCU is required to have a small resolution it may be most practical to make the smallest valves direct operated since they do not even require a powerful actuator.

An advantage with pilot operated valves is also that they have smaller electromagnetic actuators than direct operated valves. Therefore they usually require less electric power which may lead to cheaper control electronics.

There are also two important disadvantages. First is that the structure of the valve becomes more complex compared to direct operated valves. In addition to the main valve there is also the pilot valve which is usually much smaller than the main valve and therefore it often has tighter manufacturing tolerances. The flow channels are also significantly more complex which makes them more difficult to manufacture especially by drilling. This calls for new manufacturing technologies like sintering etc.

The second important disadvantage is the necessary connection between the pilot and the main stages and the leaking resulting from it. The leaking was described in more detail in chapter 5.2.1. Also in some cases it is a disadvantage that the valve requires pilot pressure to operate.

Chapter 8

Summary

This thesis described the design, manufacturing and testing of a miniature pilot-operated hydraulic on/off valve. The requirements for the designed valve were a 2 ms response time, 20 MPa operating pressure, 1,4 l/min flow rate at 0,5 MPa pressure difference and a physical size less than 5 cm^3. First a solenoid actuator was designed and simulated with FEM. A prototype of the actuator was built and tested and it was found to satisfy the requirements for the response time and the generated force. After that a pilot operated miniature on/off valve was designed and simulated in Simulink. It was decided not to build a prototype of a single valve but a prototype of a DFCU with four on/off valves. The main parts of the DFCU are four separate layers mounted together to form the necessary features for the magnetic circuits and the hydraulic parts of the four valves. The physical size of a single on/off valve in the designed prototype is about 4,5 cm^3.

The basic construction was proven to work in the test but it was noticed that the closing response time for the main valves was significantly longer than the opening response time. Also the maximum operating pressure of the valve was measured to be less than the 20 MPa required. It was determined that the return springs of the poppet of the main valve were too weak. Flow capacity of a single valve was measured to be from 1,0 to 1,4 l/min with a valve opening of 0,4 to 0,8 mm and a pressure difference of 0,5 MPa.

The DFCU prototype was modified by changing two of the four return springs to stiffer coil springs and the two others to disc spring stacks to provide more closing force for the valves. After the modification the maximum operating pressure of the DFCU exceeded 25 MPa. Also the response time was significantly improved compared to the first tests. For the valves with the disc springs the response time was about 1,5 ms at an opening of 0,4 mm and the response time was fairly independent of the pressure difference over the valve. Also with a 0,6 mm opening the closing and opening response times are less than the 2 ms design goal.

The first prototype of the designed valve satisfied the requirements for the operating pressure and physical size. Since the properties of the valve depend on the adjustable valve opening, the response time and flow rate requirements were also satisfied, but not with the same settings. However, with a 0,6 mm opening the response time with the required operating pressure is less than 2 ms and the flow rate with 0,5 MPa pressure difference is 1,2 l/min, which is 86 % of the required flow rate. Therefore it can be concluded that for a first prototype the valve can be considered quite successful.

There are still several things to improve such as leaking problems and the imbalance between the opening and closing response times. However this thesis has presented a concept which proves that pilot operation can be successfully implemented in a DFCU and it enables further miniaturization of the valves. Pilot operation enables the controlling of a larger power with smaller actuators than in direct operated valves but future applications will show if the benefits outweigh the drawbacks.

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Appendix A

Layers of the DFCU prototype

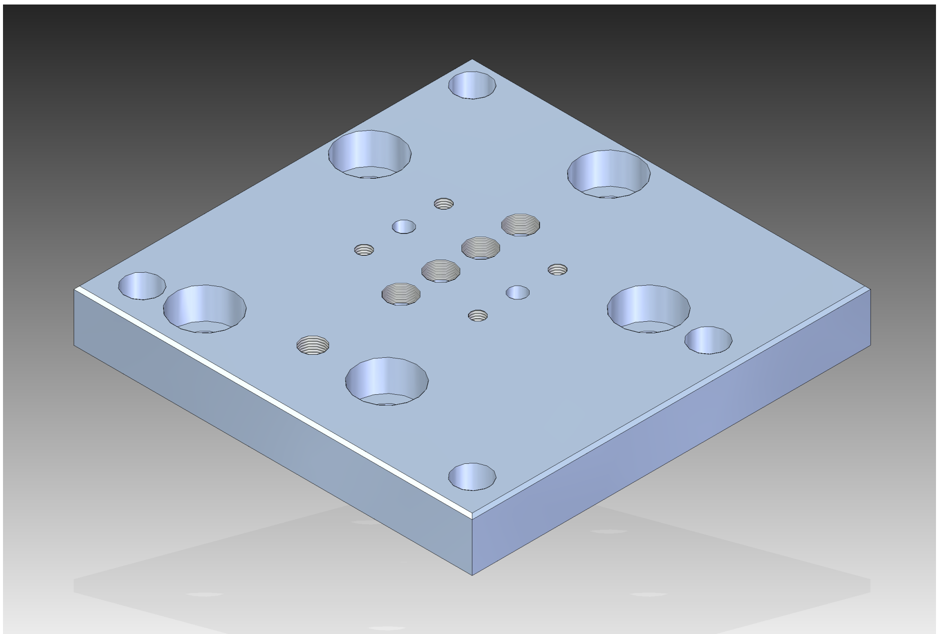

Figure A.1: Upper side of the top layer.

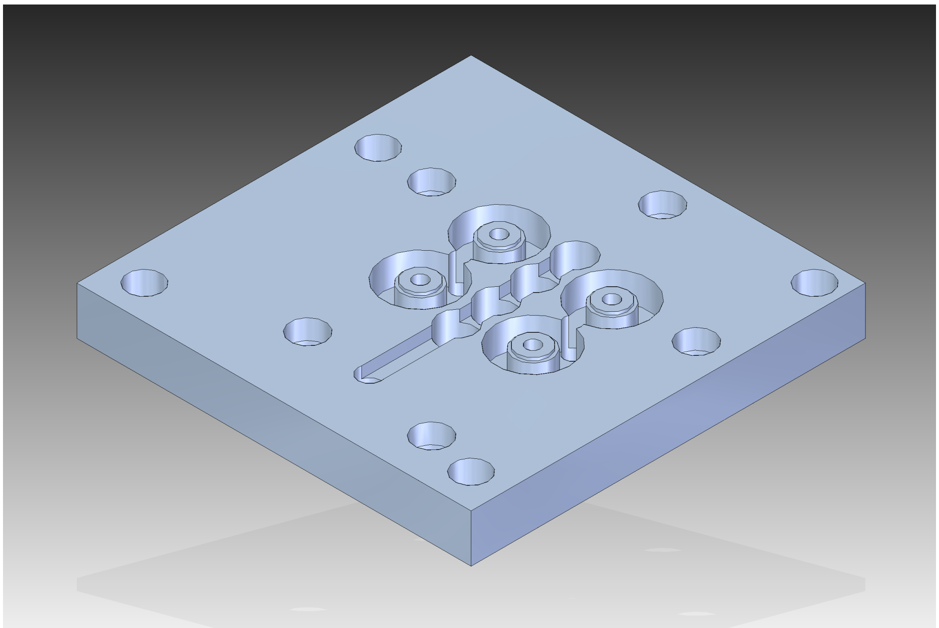

Figure A.2: Lower side of the top layer.

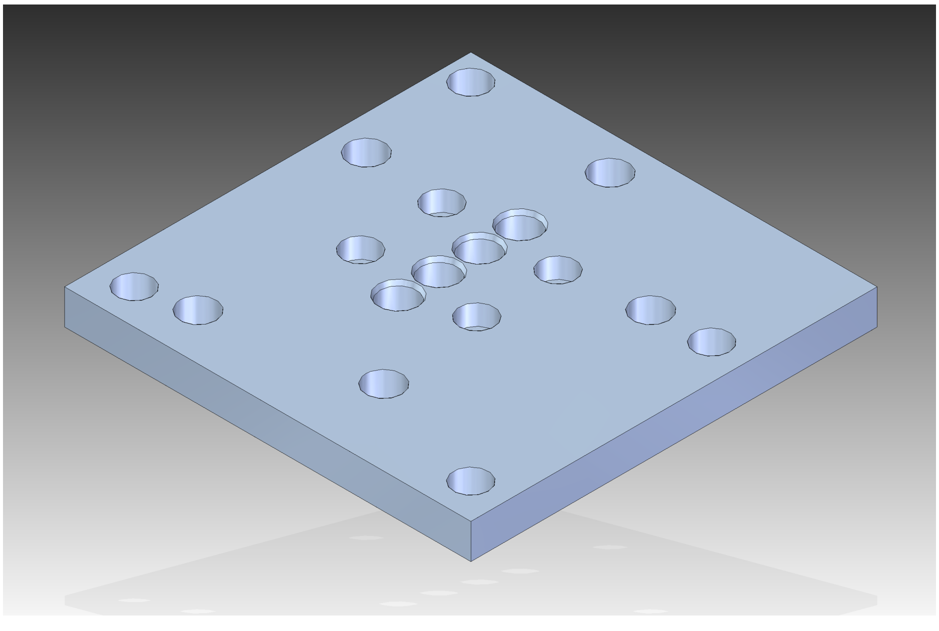

Figure A.3: Second layer from the top.

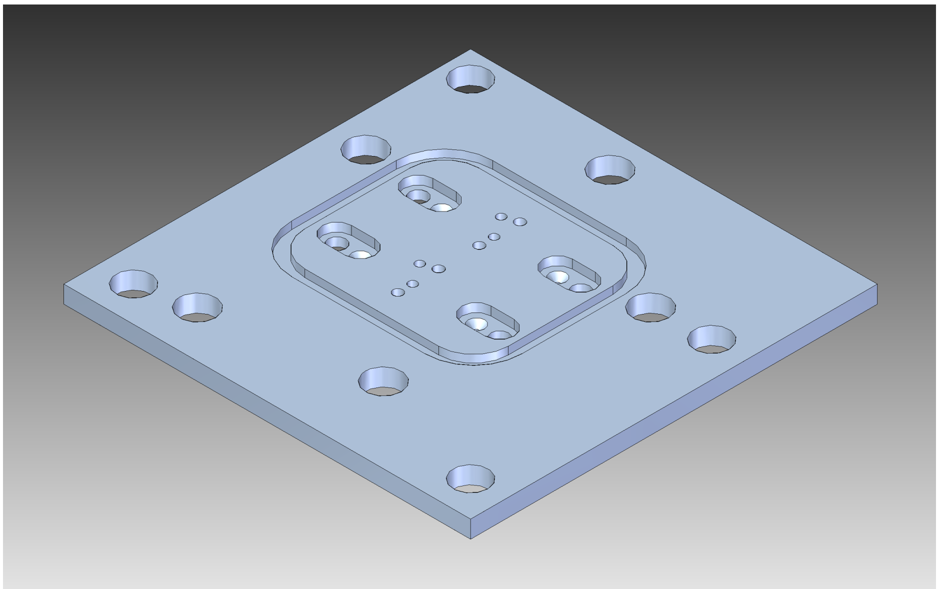

Figure A.4: Third layer from the top.

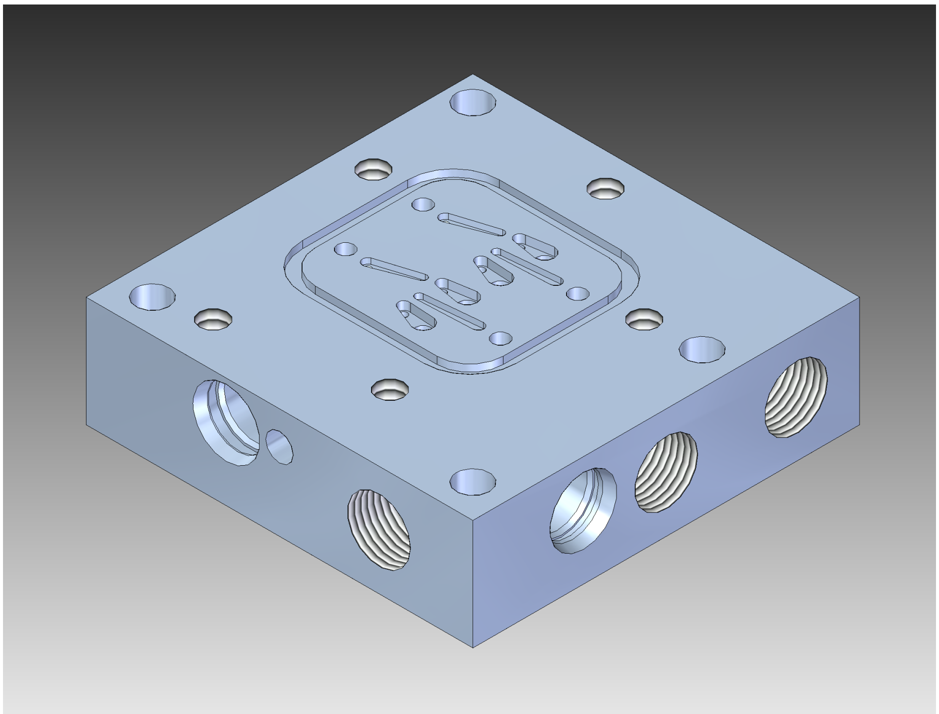

Figure A.5: Bottom block.